

101
COMPUTER MODELING OF THE HEAT
TRANSFER IN A POWERSHIFT TRANSMISSION
CLUTCH UNDER SLIPPAGE

by

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CHAPTER I

INTRODUCTION

Powershift transmissions are used in off-road vehicles and can be shifted "on-the-go" to different gear (speed) ratios while transmitting full power. When a shift is initiated by the vehicle operator, clutches in the transmission must slip momentarily in order to provide a smooth transition from one vehicle speed to another. This is necessary for operator comfort as well as to minimize shock loads on the vehicle. A powershift transmission has two directional clutches (one for "forward" and the other for "reverse") as well as several speed clutches. The direction a vehicle is traveling depends upon which directional clutch is engaged. Vehicle speed is determined by the transmission speed ratio obtained when a given set of speed clutches are engaged. The vehicle will change speed if a different set of speed clutches are engaged. Since this shift to a new speed ratio may occur at full engine power, the clutch plates will slip for a period of time until fully engaged. Some clutch slippage is desired to provide a smooth shift transition. But slippage causes the clutch plate temperature to increase rapidly since the clutch has to absorb the engine horsepower as well as the vehicle kinetic energy. Unfortunately, the life of the clutch plates is inversely related to the temperature rise. In other words, the higher the

temperature rise, the shorter the life of the clutch plates.

This research attempts to simulate and model the heat transfer, which occurs in the clutch when the vehicle transmission clutch is being "inched". "Inching" means that the vehicle is moving slowly, or is being "inched" forwards or backwards one inch at a time. The directional clutches are slipped to permit these slow vehicle speeds at high engine speeds (for greater hydraulic control, engine output torque, and vehicle maneuverability). An example of such a vehicle is the forklift. It must be able to maneuver in close quarters with a heavy payload.

OBJECTIVE

The objective of the research presented in this thesis was to develop and experimentally verify a transient heat transfer model which predicts the temperature, as a function of time, in the reaction clutch plates of a powershift transmission when "inching" takes place at a constant rate of energy dissipation (heat input). In order to accomplish this objective, it was necessary to develop a suitable experimental test setup.

CHAPTER II

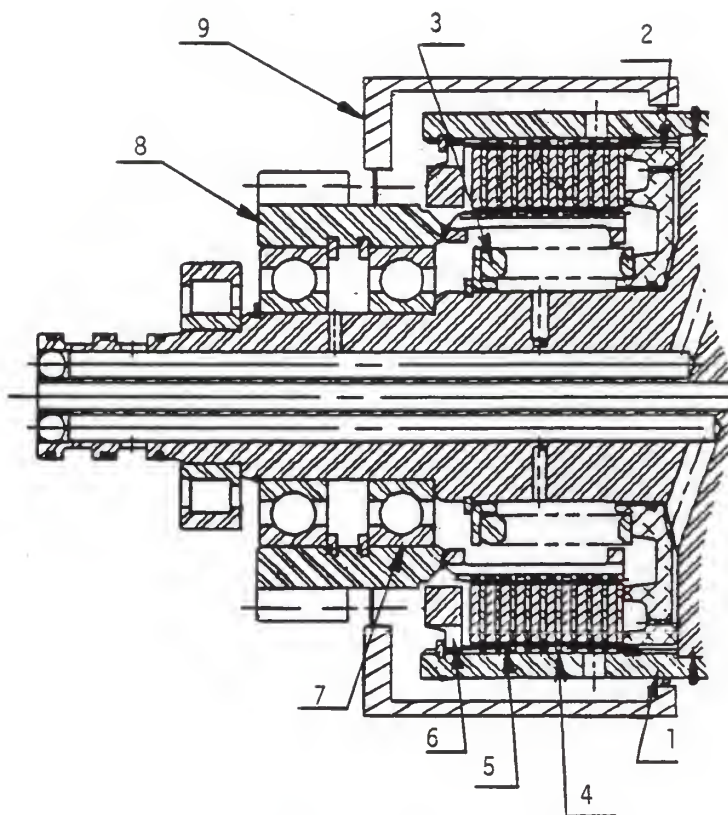
APPROACH

The approach used in this research was to first derive a heat transfer model for the type of clutch used in powershift transmissions. Next, an experimental test setup was constructed. Then, the oil flow rate through the clutch (the oil is for cooling purposes) and the reaction clutch plate temperatures were measured for various operating conditions and heat transfer coefficients were calculated using the experimental data. Finally, the heat transfer coefficients and heat input data were substituted into a "lumped parameter" transient model. The reaction clutch plate temperatures were then computed and compared to the experimentally measured reaction clutch plate temperatures in order to evaluate the accuracy of the transient model.

DEVELOPMENT OF CLUTCH HEAT TRANSFER MODEL

Description of Clutch

Figure 1 is a cross section of the clutch and shaft assembly which includes a specially fabricated shroud to collect oil from the clutch mounted in place. Use of a sealed bearing and the shroud assembly isolate the oil flowing through the forward directional clutch and direct the oil out of the transmission housing to a flowmeter for



1. Clutch Shaft Assembly
2. Clutch Piston
3. Clutch Piston Return Spring
4. Reaction Clutch Plate
5. Friction Clutch Plate
6. Clutch Plate Retainer
7. Bearing (sealed or open)
8. Forward Gear
9. Shroud (added for testing purposes)

Figure 1. Cross Section of Shaft Assembly

flow rate measurement. The clutch contains ten steel reaction clutch plates and ten composite friction clutch plates. A pattern of grooves in the surface of the friction clutch plates affects the oil flow between the clutch plates, and thus the cooling of the clutch plates. In an open clutch (a disengaged clutch), the friction clutch plates spin freely along with the reaction clutch plates. In a locked up clutch, the clutch piston clamps the clutch plates together (there is no relative velocity between friction clutch plates and reaction clutch plates). This clamping force is the supply oil pressure acting on the piston area minus the return spring force. Frictional heat is generated within the clutch when the clutch plates are allowed to slip (now there is relative velocity between the clutch plates) and when clamping force is applied to the clutch by the clutch piston. The clutch is engaged by moving the directional spool in the valve body from the neutral position to the forward position. The oil pressure behind the clutch piston (which causes the clamping force) can be controlled by positioning the "inching" spool valve at the proper location.

Lumped Clutch Analysis

In order to correctly develop a lumped parameter clutch heat transfer model, the Fourier number and Biot number must

meet certain requirements.

Figure 2 is a sketch of one of the steel reaction clutch plates and one of the composite friction clutch plates for a typical powershift transmission clutch. Since the friction material contains asbestos, it was assumed that the friction material is an insulator, and that the heat entering any of the composite clutch plates can be neglected. Two reaction clutch plate thicknesses were used during this research, "thin" plates measuring 0.070 in. and "thick" plates measuring 0.125 in.

The Fourier number (Fo) is the ratio of the heat conduction rate to the rate of thermal energy storage in a solid. For a step or constant heat input it is defined as:

$$Fo = a t / L^2 \quad (1)$$

where:

a = reaction clutch plate thermal diffusivity, ft^2/hr

t = clutch slip time to steady state plate temperature, sec

L = reaction clutch plate half thickness, in

As will be seen later in this thesis, the reaction clutch plate temperatures were measured using thermocouples imbedded in the "thick" reaction plates. The clutch slip time to reach steady state reaction plate temperature was typically no less than 8 seconds. These conditions result in a Fourier number of 38.

Some researchers have studied the case of heat input into a clutch through a rotating flywheel [1]. In this case, the

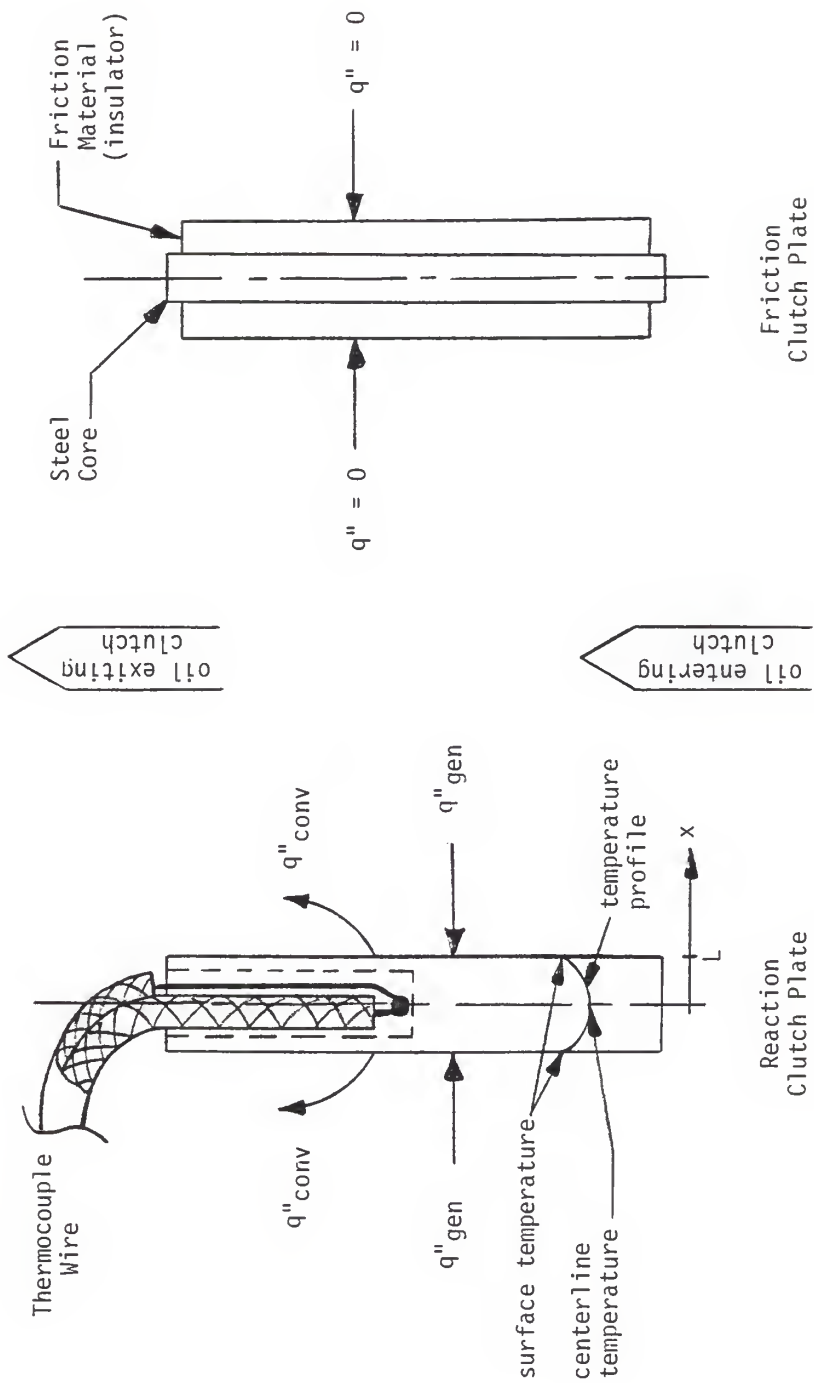


Figure 2. Schematic of Reaction and Friction Clutch Plates

reaction clutch plate surface temperature is considered equal to the reaction clutch plate centerline temperature when the Fourier number is greater than 3. Although this is not the same case as in this research, it does provide a basis for comparison.

The Biot number provides a measure of the temperature drop in the reaction clutch plate relative to the temperature difference between the surface and the fluid. If $Bi \ll 1$, the resistance to conduction within the reaction clutch plate is much less than the resistance to convection across the boundary layer [2]. The Biot number (Bi) is defined as:

$$Bi = H L / k \quad (2)$$

where:

H = heat transfer coefficient, $\text{btu/hr-in}^2\text{-F}$

L = reaction clutch plate half thickness, in

k = reaction clutch plate thermal conductivity, btu/hr/ft/F

If the reaction plate is to be considered to have a uniform temperature (no temperature gradient in the x direction) during the clutch slip time, then $Bi \leq 0.1$. As will be seen later in this thesis, the heat transfer coefficient values encountered during this research were typically around 3 to 4. These results give Biot numbers of 0.09 and 0.12, thus use of a lumped model is justified. This is an important point because the convective heat transfer occurs due to the reaction clutch plate surface

temperature and not the reaction clutch plate centerline temperature (see Figure 2). The reaction clutch plate temperature values used in all the calculations within this thesis are centerline temperatures.

A special computer program, using the forward difference method, was used to calculate reaction clutch plate surface temperatures and reaction clutch plate centerline temperatures (see Figure 2). Experimental test data was used to calculate the heat input into the clutch. It was assumed that all the heat was absorbed by the reaction clutch plates, and that no heat was absorbed by the oil flowing through the clutch or the friction clutch plates. These assumptions resulted in the maximum possible reaction clutch plate temperature using the given heat input. The difference obtained between the surface temperature and the centerline temperature was no greater than 20 F during the temperature transient.

Formulation of Instantaneous Clutch Heat Transfer Model

An instantaneous heat balance around the clutch, assuming a lumped parameter model, gives the equation:

$$\dot{m} [C_p (T_{OUT}) T_{OUT} - C_p (T_{IN}) T_{IN}] = H A K (T_{PLATE} - T_{MEAN}) \quad (3)$$

where:

A = total reaction clutch plate surface area, in²
C_p(T) = specific heat of oil at temperature T, btu/lb/F
GPM = volumetric oil flow rate through clutch, gal/min
H = heat transfer coefficient, btu/hr-in²-F
K = conversion factor, 60
.
m = GPM * RHO(T) * 231 / 1728
= mass oil flow rate through clutch, lb/min
RHO(T) = density of oil at temperature T, lb/ft³
T_{IN} = temperature of oil entering clutch, F
T_{OUT} = temperature of oil exiting clutch, F
T_{MEAN} = .5 (T_{IN} + T_{OUT})
= arithmetic mean oil temperature, F
T_{PLATE} = reaction clutch plate temperature, F

Substituting for T_{mean} and solving for the heat transfer coefficient H results in Equation 4 below:

H = 8.0208 *

$$\frac{\text{GPM} [\text{RHO}(\text{T}_{\text{OUT}}) \text{C}_p(\text{T}_{\text{OUT}}) \text{T}_{\text{OUT}} - \text{RHO}(\text{T}_{\text{IN}}) \text{C}_p(\text{T}_{\text{IN}}) \text{T}_{\text{IN}}]}{\text{A} [\text{T}_{\text{PLATE}} - .5 (\text{T}_{\text{OUT}} + \text{T}_{\text{IN}})]} \quad (4)$$

It should be noted that H is actually an average heat transfer coefficient for the entire clutch. All the temperature terms in Equation 4 are experimental data points taken at specific points in time. The oil flow rate, oil density and oil specific heat are calculated from other instantaneous measurement values. These "instantaneous" terms were used to calculate experimental values for the heat transfer coefficient H.

CHAPTER III

TEST EQUIPMENT

Transmission

A Funk 2133 series powershift transmission was used in this research. It is a 2000 series "short drop" transmission with three speeds in both forward and reverse. The forward directional clutch was used for all the tests conducted during this research. The friction clutch plates used during this research had asbestos friction material with grooves in the friction material to allow flow of oil for cooling and lubricating purposes. A stall bar was connected to the transmission output shaft to prevent the reaction clutch plates of the forward directional clutch from turning. This was necessary since thermocouples were mounted in some of the reaction clutch plates.

Engine

The engine used to supply power to the transmission was a Caterpillar model 3204 DI-T. This 4 cylinder turbocharged diesel engine has a maximum horsepower of approximately 120 HP. The high idle speed was set at 2650 RPM.

Instrumentation

The following engine and transmission functions were monitored:

- 1) Engine speed: Airpax magnetic pickup, Hoffer ACC-7B signal conditioner
- 2) Transmission input gear speed: Airpax magnetic pickup, Daytronics model 9141 horsepower module, 0-3000 RPM, 0-5 VDC
- 3) Transmission output torque: Sensotec model 41/572-05 load cell, Daytronics model 9170 strain gage conditioner 0-5000 FT-LB, 0-5 VDC, 2 HZ low pass filter
- 4) Forward directional clutch oil pressure: Pace Wiancko model P7D-10PSID-785 transducer, Pace Wiancko model CD10 carrier demodulator, 0-300 PSI, 0-5 VDC
- 5) Lube in oil pressure: Sensotec model A 10/1036-17 transducer, 0-100 PSI, 0-5 VDC
- 6) Pump oil flow rate: Potter model 435 turbine flowmeter, Hoffer ACC-7B signal conditioner
- 7) Lube in oil flow rate: Potter model 429 turbine flowmeter, Hoffer ACC-7B signal conditioner
- 8) Forward directional clutch oil flow rate: Hoffer model HO1/2X1/4-.1-4.5-B-1MC3P-MS turbine flowmeter, Hoffer ACC-27 modulated carrier conditioner/converter, Daytronics model 840 frequency to voltage converter, 0-1500 HZ, 0-5 VDC
- 9) Lube in oil temperature: type T thermocouple, Transmation model 3610T transmitter, 0-250 F, 0-5 VDC
- 10) Forward directional clutch flowmeter temperature: type K thermocouple, Transmation model 3610T transmitter 0-500 F, 0-5 VDC
- 11) Clutch reaction plate temperatures (8 locations): type K thermocouples, Transmation model 3610T isolated transmitters 0-1500 F, 0-5 VDC

Data Acquisition System and Software

A Zenith 151 PC with one hard disk drive and one floppy disk drive was used for data acquisition and analysis. The engine and transmission instrumentation was interfaced to

the Zenith by a MetraByte DASH-16 expansion board and a MetraByte CTM-05 expansion board. The DASH-16 is a multi-function high speed analog/digital I/O expansion board. It was used to sample 15 conditioned 0 - 5 VDC signals at an A/D clock frequency of 150 HZ (one scan through the 15 inputs every 0.1 seconds). The CTM-05 is a multi-function counter-timer and digital I/O expansion board. It was used to measure engine speed and flowmeter frequencies by counting the signal pulses over a period of time.

The software developed for this research permitted monitoring engine/transmission functions, acquiring data and analyzing data. The programming was done in Microsoft GW-BASIC, and followed the MetraByte DASH-16 and CTM-05 programming manuals.

CHAPTER IV

PRELIMINARY TESTING

Purpose of Preliminary Testing

As previously stated, the objective of this research was to develop a transient heat transfer model for the clutch of a powershift transmission. This model requires using an average heat transfer coefficient H as defined by Equation 4. According to Equation 4, H then is a function of the oil flow rate through the clutch, reaction plate temperatures, specific heat of the oil, density of the oil and clutch plate surface area. Thus the purpose of the preliminary testing was to determine oil flow rates through the clutch as well as to determine temperature variations within the clutch for different engine and transmission operating conditions. These tests were necessary in order to better understand how the clutch behaves, and also to discover any complications affecting a measurement of the variables needed for calculation of the heat transfer coefficient H .

Measurement of Oil Flow Rate

Direct Method

The oil flow rate through the forward directional clutch was first determined by direct measurement for various operating conditions. These conditions are listed

below.

- 1) Engine speed
- 2) Friction clutch plate speed
- 3) Initial stackup clearance between all of the clutch plates
- 4) Pressure behind clutch piston (open and closed clutches)
- 5) Temperature of oil entering clutch
- 6) Temperature of reaction clutch plates

The temperature of the oil entering the clutch was held between 170 F to 180 F during all tests within this research. Figure 3 shows the oil flow rate through an open (zero clutch piston pressure) clutch. The transmission output shaft was locked up by using a stall bar to prevent the transmission cylinder gears from turning. Thus the reaction clutch plates were stationary and only the friction plates were rotating.

Figure 3 shows the total oil flow possible through the open clutch and was obtained using a sealed bearing installed to prevent oil, from the clutch, from flowing through the bearing (see Figure 1). Thus all oil exiting the four lubricant orifices in the clutch shaft must pass through the clutch. The oil flow was collected by installing a special shroud (see Plate 1) around the clutch cylinder. A 1" I.D. hose connected the shroud to an electric pump which evacuated the oil from the shroud

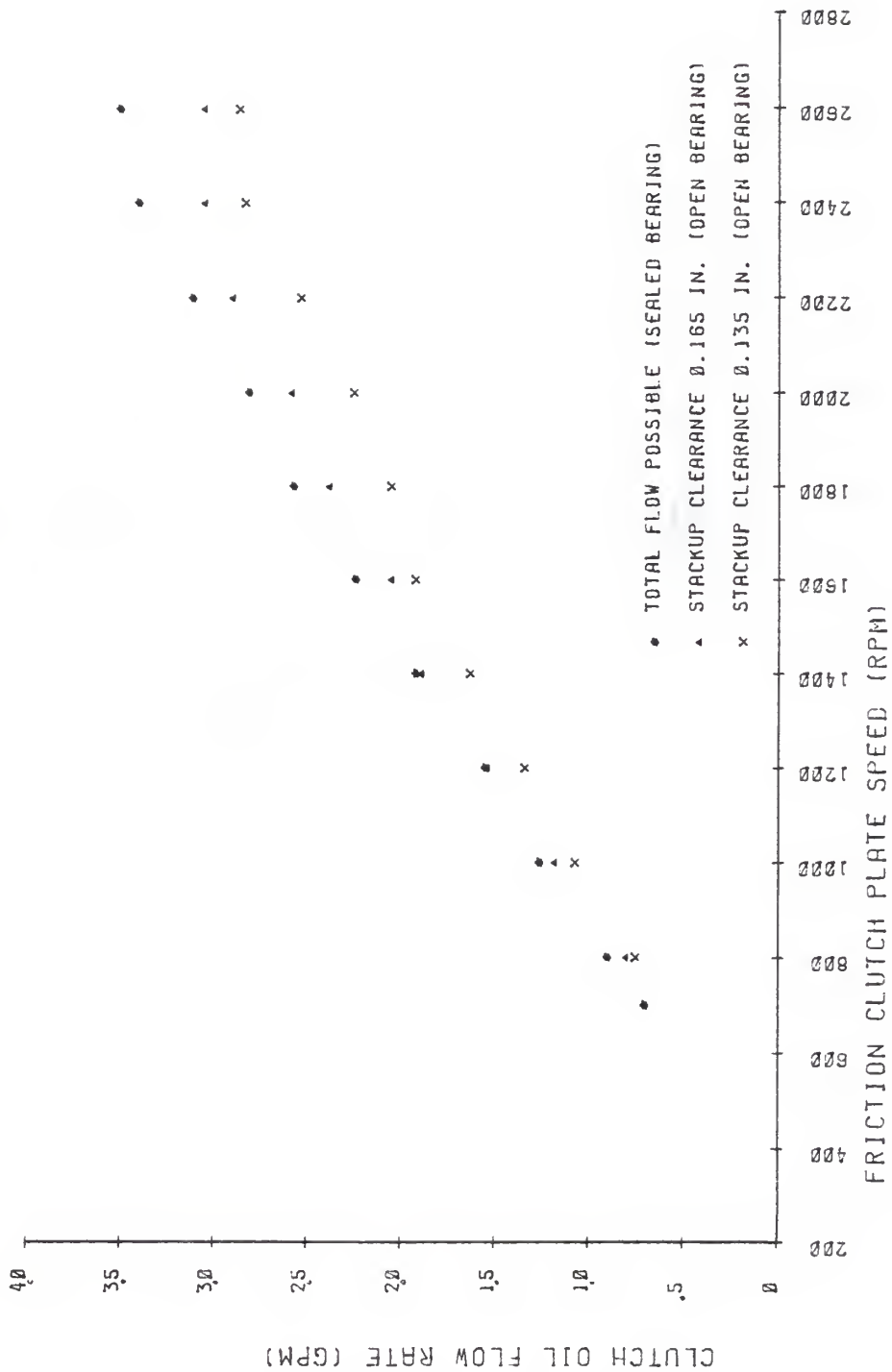


FIGURE 3. OIL FLOW RATE THROUGH OPEN CLUTCH



Plate 1. Shroud



Plate 2. Oil Exit Temperature Thermocouple
in Hole

and pumped it into a container. The time and volume of oil to fill the container were measured in order to calculate the oil flow rate through the clutch. As expected, the clutch oil flow rate increased with increasing friction clutch plate speed.

The two lower curves in Figure 3 show the effect that clutch plate stack-up clearance has upon oil flow rate. Note that these two curves are for clutch shafts using an open bearing. This was done to maintain the transmission as designed (the transmission does not normally use any sealed bearings).

The oil flow rate measurements made by the direct method are compared to the oil flow rate measurements obtained using the turbine flowmeter in the next two sections of this thesis. This was done to verify the accuracy of both methods of oil flow rate measurement.

Turbine Flowmeter

A turbine flowmeter was next installed in the shroud oil exit line and the flow rate test to determine the total flow possible through the clutch was repeated. These results are also plotted in Figure 3.

Figure 4 shows the oil flow rate through the friction clutch plate grooves in a locked up clutch. The oil flow rate through the clutch is proportional to engine speed

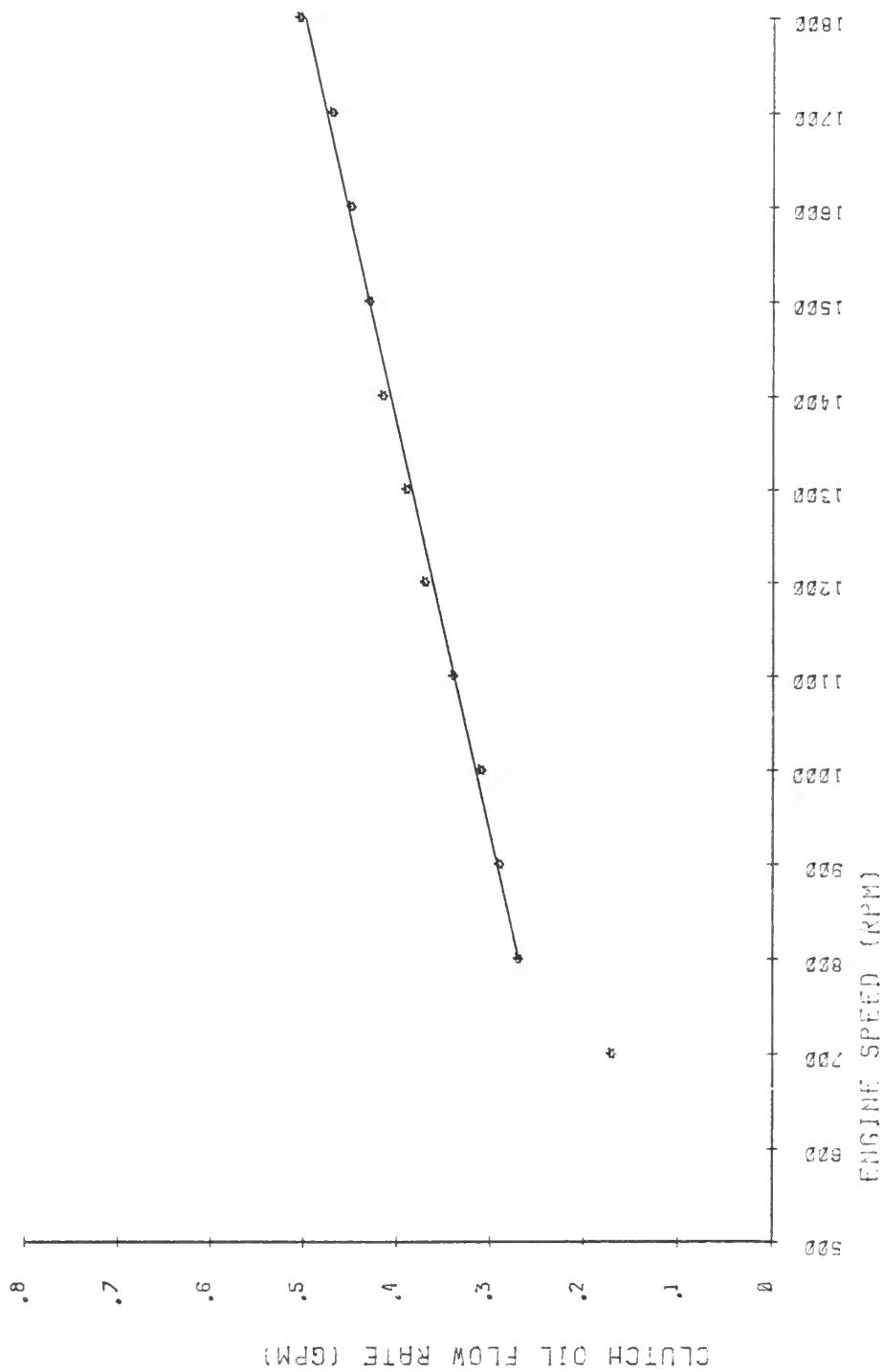


FIGURE 4. OIL FLOW RATE THROUGH CLOSED CLUTCH

because the oil pump is driven by the engine. The transmission output shaft is locked up as in all testing but now the torque converter is stalled. None of the clutch plates are slipping or rotating, therefore the only flow possible is through the grooves in the friction clutch plates. Hydraulic pressure filling the cavity inside the clutch forces the oil to flow through these clutch plate grooves. These results show the amount of oil flow through the friction clutch plate grooves alone. Results discussed later in the thesis will show the total amount of oil flow through all of the clutch plates. Combining these two results will show the amount of oil flow over the reaction clutch plate surfaces versus the amount of oil flow through the friction clutch plate grooves.

Summary of Preliminary Oil Flow Rate Testing

No significant change in oil flow rate was detected between the direct method of flow rate measurement and the turbine flowmeter method of flow rate measurement, even though addition of the flowmeter increased the resistance to flow in the oil exit line from the clutch. The oil flow rates shown in Figure 3 were measured both directly and by the turbine flowmeter, both methods giving essentially the same results.

Measurement of Oil and Reaction Clutch Plate Temperatures

Installation of Thermocouples

The temperature of the steel reaction clutch plates and the temperature of the oil exiting the clutch were measured using type "K" chromel-alumel thermocouple wires inserted into "thick" and "thin" reaction plates, respectively (see Appendix A for all material properties and dimensional specifications). Reaction clutch plate temperatures were obtained by drilling a hole through the circumference of the plate to the center line of the plate (see Figure 5). The thermocouple wires were first spot welded together and then held in place within the drilled hole by ceramic cement. The glass braid thermocouple wire insulation was previously treated with epoxy to prevent the insulation ends from fraying while being handled. The insulation of the alumel lead was trimmed back to allow insertion of the complete chromel lead (wire plus insulation) and the alumel lead (wire without insulation) into the drilled hole. This was done to electrically isolate the thermocouple leads from each other, even though the thermocouple transmitters are already isolated units. The thermocouple leads were threaded along the clutch cylinder and out the shroud as shown in Plate 5. Two spline teeth were ground off the reaction plates at each of the six thermocouple positions to compensate for

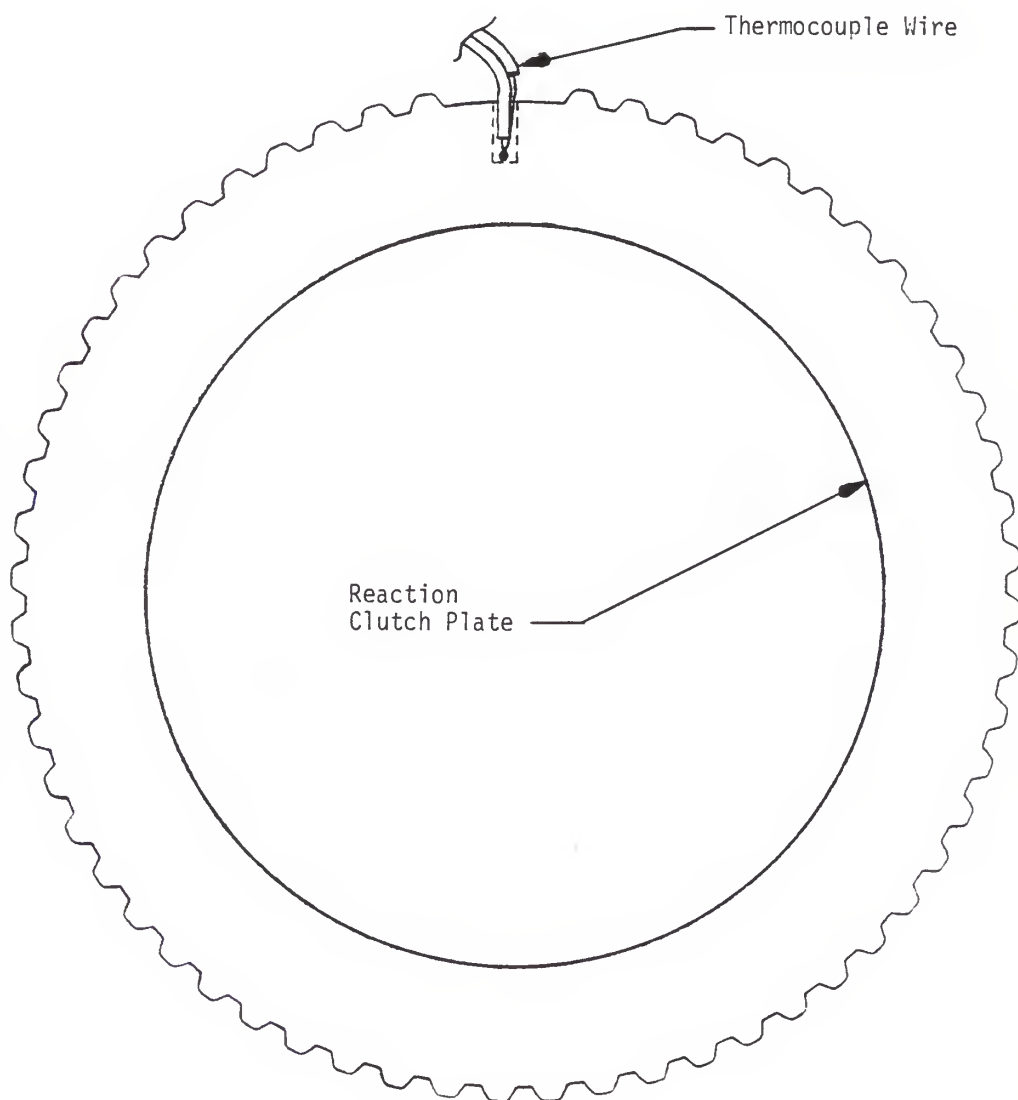
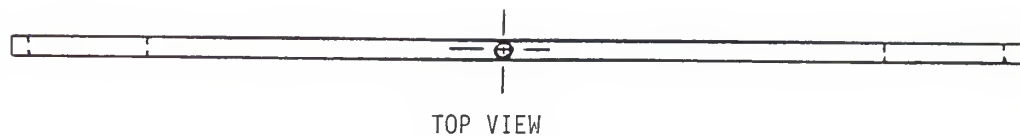


Figure 5. Thermocouple Placement for Reaction
Clutch Plate Temperature Measurement

the flow restriction caused by the thermocouple wires.

The temperature of the oil exiting the clutch was measured near the outer edge of the reaction clutch plate. A hole was drilled through the circumference of the plate towards the center of the plate. The thermocouple was again inserted into the drilled hole, but first a hole was drilled through the reaction clutch plate perpendicular to the plate surface (see Plate 2). One side of the hole was beveled to promote oil flow through the hole in the plate. This hole is located near the outer edge of the reaction plate. The thermocouple junction is located at the center of the hole, but one of the thermocouple wires extends past the hole further into the plate. This serves to support the thermocouple wires and also to isolate the thermocouple junction at the center of the drilled hole and away from any plate temperature effects. A second method of oil temperature measurement was also attempted by cutting a recess in the reaction clutch plate outer circumference. The thermocouple junction was then located within this recess (see Plate 3).

The drilled hole method of oil exit temperature measurement is the preferred method of temperature measurement (for reasons discussed later) and is therefore used for all such measurements within this research.



Plate 3. Oil Exit Temperature Thermocouple
in Slot



Plate 4. Reaction Clutch Plate for Temperature Measurement

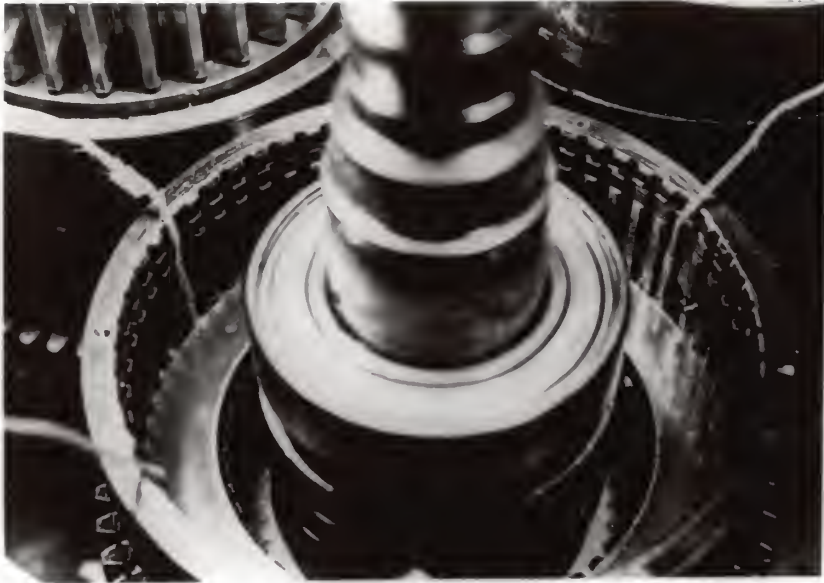


Plate 5. Thermocouple Leads in Cylinder Wall Grooves



Plate 6. Axial Hole Locations in Clutch Cylinders

Effect of Shroud and Flowmeter on Reaction Clutch Plate Temperature Measurements

The shroud was placed around the clutch during some of the temperature tests to collect the oil for flow rate measurement. Insertion of the turbine flowmeter into the system causes a restriction in the oil exit line, as mentioned previously. It was expected that this restriction would reduce the oil flow rate through the clutch, therefore causing higher reaction clutch plate temperatures. In order to determine whether the oil flow restriction was a significant factor, the flowmeter was replaced by a pump connected to the shroud to evacuate the oil from the shroud (this is the direct method of oil flow rate measurement discussed earlier). No significant reaction clutch plate temperature differences were noticed between these two tests, therefore it was concluded that the reaction clutch plate temperatures are not adversely affected by the insertion of the flowmeter into the oil exit line.

Effect of Circumferential Location of Thermocouples on Reaction Clutch Plate Temperature Measurements

The purpose of this test was to determine if the reaction clutch plate temperature varied in the circumferential direction under constant clutch slip conditions. A reaction clutch plate with six thermocouple

wires, as shown in Figure 6 and Plate 4, was placed in the center (reaction clutch plate 5) of the forward directional clutch pack. The clutch normally contains 10 reaction plates and 10 friction plates, with plate 5 located at the center of the clutch pack. Figure 7 shows typical reaction clutch plate temperature rises measured around the plate circumference. Temperature rise is the increase in reaction clutch plate temperature caused by a heat input at the plate surface. Mathematically, the temperature rise is the final reaction clutch plate temperature at clutch slip time t minus the initial reaction clutch plate temperature at time zero. The temperature rise is used to prevent the temperature of the oil entering the clutch (which can fluctuate during the tests) from affecting the final reaction clutch plate temperature results. The lower temperature rises shown in Figure 7 correspond to those thermocouples near an oil exit hole in the clutch cylinder. Similarly, the higher temperature rises correspond to those thermocouple locations further from an oil exit hole. There are 6 oil exit holes drilled through the clutch cylinder wall (1 every 60 degrees), and the holes have three different axial locations (see Plate 6). These holes permit the oil to exit the clutch cylinder and return to the transmission sump (or to be collected by the shroud, as was the case for most of the tests during this research). Thus the

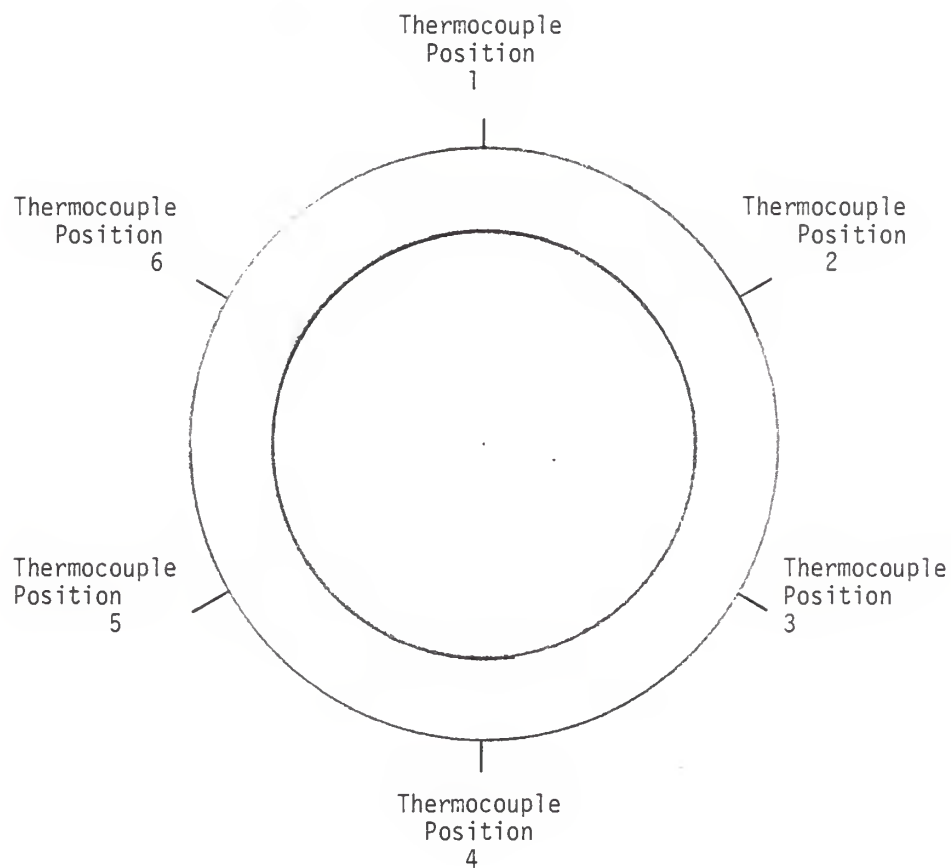


Figure 6. Thermocouple Placement Around Reaction Clutch Plate Circumference

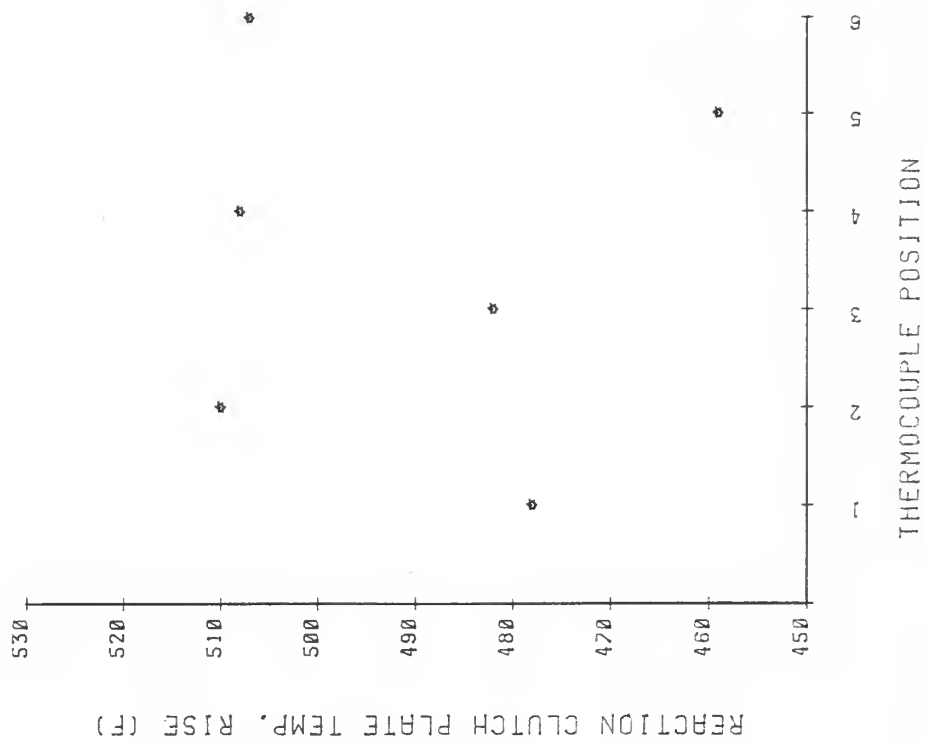


FIGURE 7. CIRCUMFERENTIAL REACTION CLUTCH PLATE TEMPERATURE DISTRIBUTION

variation of plate temperature rise around the plate circumference is determined by the location of hole placement in the clutch cylinder wall. This observation has significant impact on where the reaction clutch plate temperature rise measurements should be made to obtain an average temperature rise to be used in heat transfer coefficient calculations. The heat transfer coefficient value could be either high or low depending upon the location chosen. As will be seen later in this thesis, the temperature rise used in the calculation of the average heat transfer coefficient was the average of six thermocouple measurements equally spaced around the circumference of one reaction clutch plate. Plate 4 shows the six thermocouple wires leading from this reaction clutch plate.

Effect of Axial Location of Thermocouples on Reaction Clutch Plate Temperature Measurements

The reaction clutch plate temperature rise also varies axially through the clutch pack due to the same reasons stated above (location of oil exit holes in clutch cylinder). Sample results are shown in Figure 8. One investigator [4] has shown that clutch plate spline friction reduces the force applied to each succeeding plate, moving away from the piston. Thus reaction plate 10 (next to the retainer) should have less heat input than

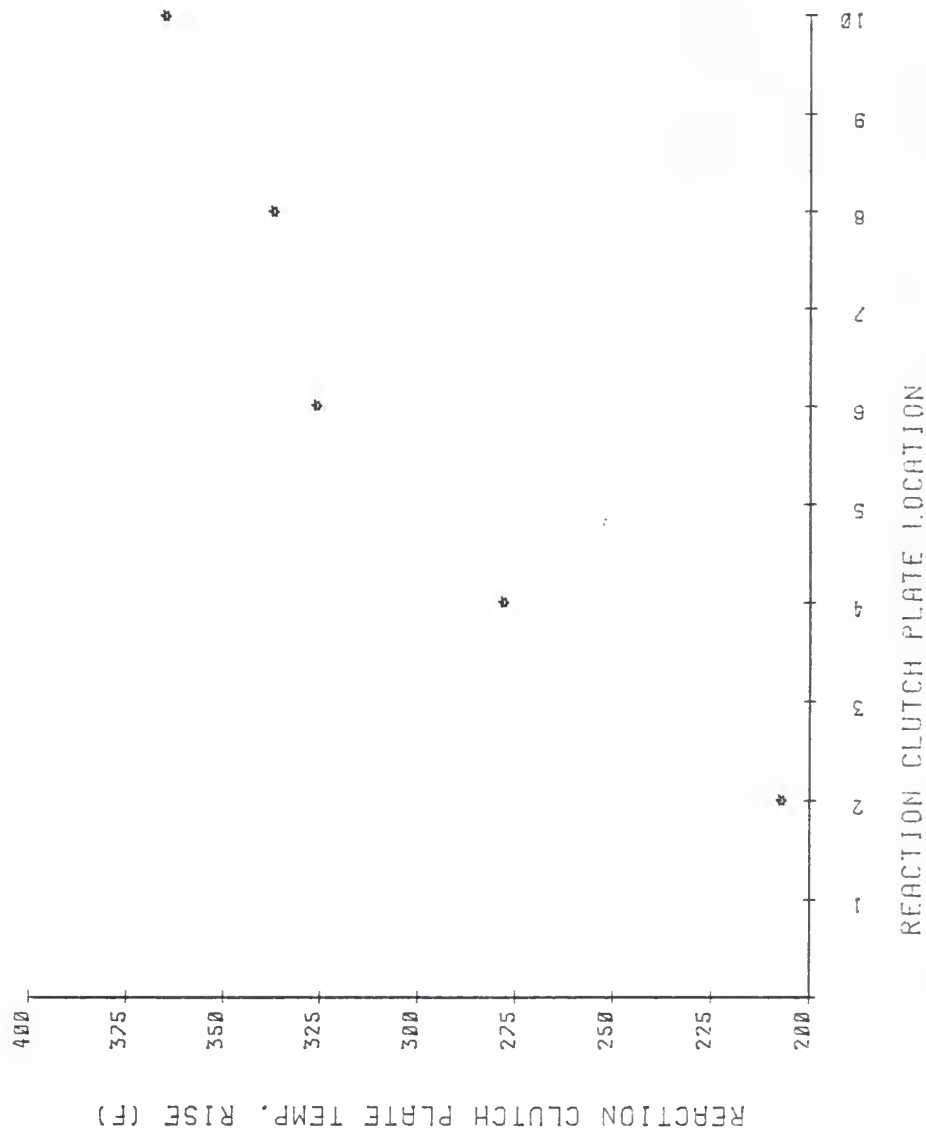


FIGURE 8. AXIAL REACTION CLUTCH PLATE TEMPERATURE DISTRIBUTION

plate 1 (next to the piston). But in Figure 8, reaction plate 2 has the minimum temperature. It was concluded that the oil flow through the clutch must be greater here than elsewhere in order to maintain this lower plate temperature. It was not determined whether spline friction is a significant factor in the clutch tested.

The maximum reaction clutch plate temperature difference can be defined as:

$$\Delta T_{MAX} = \frac{T_{PLATE} - T_{AVG}}{T_{AVG}} \quad (5)$$

where:

T_{PLATE} = maximum measured reaction clutch plate temperature, F

T_{AVG} = average of the axial reaction clutch plate temperature measurements, F

Typically, the maximum clutch plate temperature difference is 14%. Similarly, the minimum reaction plate temperature difference, ΔT_{min} , can also be defined and is typically 20%. Thus the axial temperature variations within the clutch are significant and must be considered along with the circumferential temperature variations.

Figure 9 shows the axial plate temperature distribution at nine reaction clutch plate positions. These positions are the same as those in Figure 8. In order for the oil to exit the clutch through the oil exit holes in the cylinder

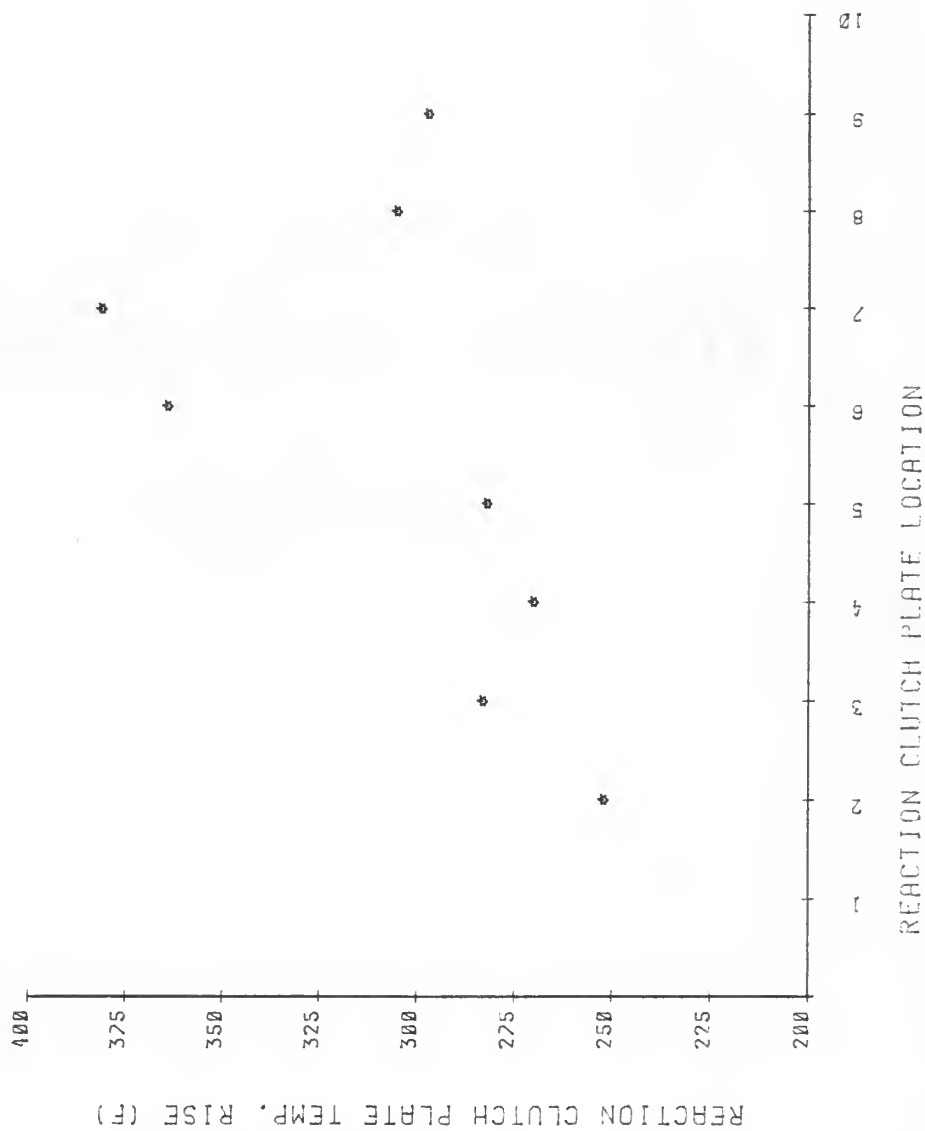


FIGURE 9. AXIAL REACTION CLUTCH PLATE TEMPERATURE DISTRIBUTION

wall, the oil must flow along the cylinder wall through the slots cut in the wall (see Plate 5). Unfortunately, the axial oil flow rate along this side of the cylinder wall, and out the corresponding oil exit holes, was reduced significantly because the nine thermocouple wires restricted the oil passageway. Thus the data in Figure 9 probably contains some error, however it does have the same trend as the results shown in Figure 8.

Reaction clutch plate 5 was selected as the "best" axial location for the reaction clutch plate temperature rise measurements used to calculate values for the average heat transfer coefficient.

Effect of Radial Location of Thermocouples on Reaction Clutch Plate Temperature Measurements

The reaction clutch plate temperature rise was also expected to vary as a function of the thermocouple radial position (that is, to vary according to how deep the hole in the plate was drilled for the thermocouple). This effect was not pursued in the course of this research since the lumped model assumed requires only the average plate temperature.

Effect of Circumferential Location of Thermocouples on Oil Exit Temperature Measurements

The temperature of the oil exiting the clutch also

varies around the reaction clutch plate circumference. The oil exit temperature was measured using a "thin" reaction clutch plate located at plate position 6 (plate position 1 is next to the clutch piston). It was noticed that the oil exit temperature was equal to or greater than the plate centerline temperature when the thermocouples were placed directly in front of an oil exit hole, for example, circumferential position 1. When the thermocouples were moved to a location more distant from the oil exit holes, such as circumferential positions 2 or 3, then the plate temperature was greater than the oil exit temperature, which was expected.

The reaction clutch plate used to measure the oil exit temperature, needed to calculate the heat transfer coefficient, was rotated so that the thermocouple was located at circumferential position 2 (see Figure 6).

Effect of Axial Location of Thermocouples on Oil Exit Temperature Measurements

The temperature of the oil exiting the reaction clutch plates in the axial direction is expected to be similar to the axial clutch plate temperature variation discussed above. This effect was not pursued in the course of this research.

Reaction clutch plate 6 was the axial location chosen for the oil exit temperature measurements used in the

average heat transfer coefficient calculation.

Effect of Radial Location of Thermocouples on Oil Exit Temperature Measurements

The preferred method of measuring the temperature of the oil exiting the clutch is to use a thermocouple located within a hole drilled through the plate surface, as discussed previously (see Plate 2). This method isolates the thermocouple junction from any outside temperature affects. However, at the same time, it also causes an error in the oil exit temperature measurement. Since the hole cannot be located at the outer edge of the plate, the true temperature of the oil exiting the clutch cannot be measured, but rather an average of the oil temperature on the plate surface. Some researchers have studied how the oil flows over the surfaces of rotating clutch plates [3]. They concluded that the oil travels over smooth clutch plate surfaces in an outward spiral movement. From this, it was concluded that for this research the error in the oil exit temperature measurement is small since the path that the oil travels over the plate surface is long relative to the diameter of the hole drilled through the plate surface.

On the other hand, a thermocouple located within the recess of the reaction clutch plate, as previously mentioned (see Plate 3) may be cooled by oil flowing along

the clutch cylinder wall, or by an oil flow vortex within the recess cavity. Even so, oil exit temperature measurements by thermocouples located within the recess of the reaction clutch plate compared favorably to those using a thermocouple placed in a drilled hole in the reaction clutch plate.

The drilled hole method of oil exit temperature measurement was the method used within this research for all such oil exit temperature measurements.

Errors in Temperature Measurements

The total error in these temperature measurements is not certain. Drilling holes into the reaction clutch plates for thermocouple installation causes errors in temperature measurement. The drilling process creates a "high" spot on the plate surface because the drill bit deforms the metal around the hole. More heat is generated on the "high" spot, thereby causing high plate temperatures around the hole. The thermocouples are cemented into the drilled hole by a ceramic cement. The thermal conductivity of the cement is different than that of steel (ceramic cement has a thermal conductivity of 3.4 compared to a steel conductivity of 300) and therefore insulates the thermocouple from the reaction clutch plate. This causes a low reaction clutch plate temperature reading.

There are errors between different thermocouple measurements because each thermocouple has its respective transmitter. Although the transmitters were all calibrated at the same points, each transmitter has a calibration error of 1.5 F. An attempt to verify the accuracy of the temperature measurements and transmitters was made by switching the thermocouple leads to different transmitters, and also by rotating the reaction plate used for plate temperature measurement to a different position within the clutch cylinder. None of these changes resulted in different temperature profiles nor affected the temperatures reached at the plate positions by any large degree.

Finally, there is a quantization error due to the 12 bit A/D converter within the ZENITH computer.

CHAPTER V

TESTING TO EXPERIMENTALLY DETERMINE CLUTCH HEAT TRANSFER COEFFICIENTS

Test Setup and Procedure

Plates 7 and 8 show the engine/transmission test setup. The transmission plumbing is as specified by Funk installation drawings.

Plate 9 shows the container used to determine average clutch flow rates by direct measurement. The oil was evacuated from the clutch shroud using an auxiliary pump and pumped into the container. The container oil level was determined by reading the burette on the side of the container. The oil level in the container was controlled by a manual valve underneath the container which allowed dumping the oil from the container into the transmission sump. The sump was initially overfilled (with four gallons of oil) so that the five gallon container could be filled for volume measurement without emptying the sump. The elapsed time measurements ranged from 30 to 130 seconds.

Plate 10 shows the computer and instrumentation cabinet. Data collection was initiated at the same time the transmission clutch was engaged by flipping a switch which sends a 5 volt DC signal to the computer and 115 volts AC to a solenoid. The solenoid operates the directional spool valve in the transmission valve body to shift the

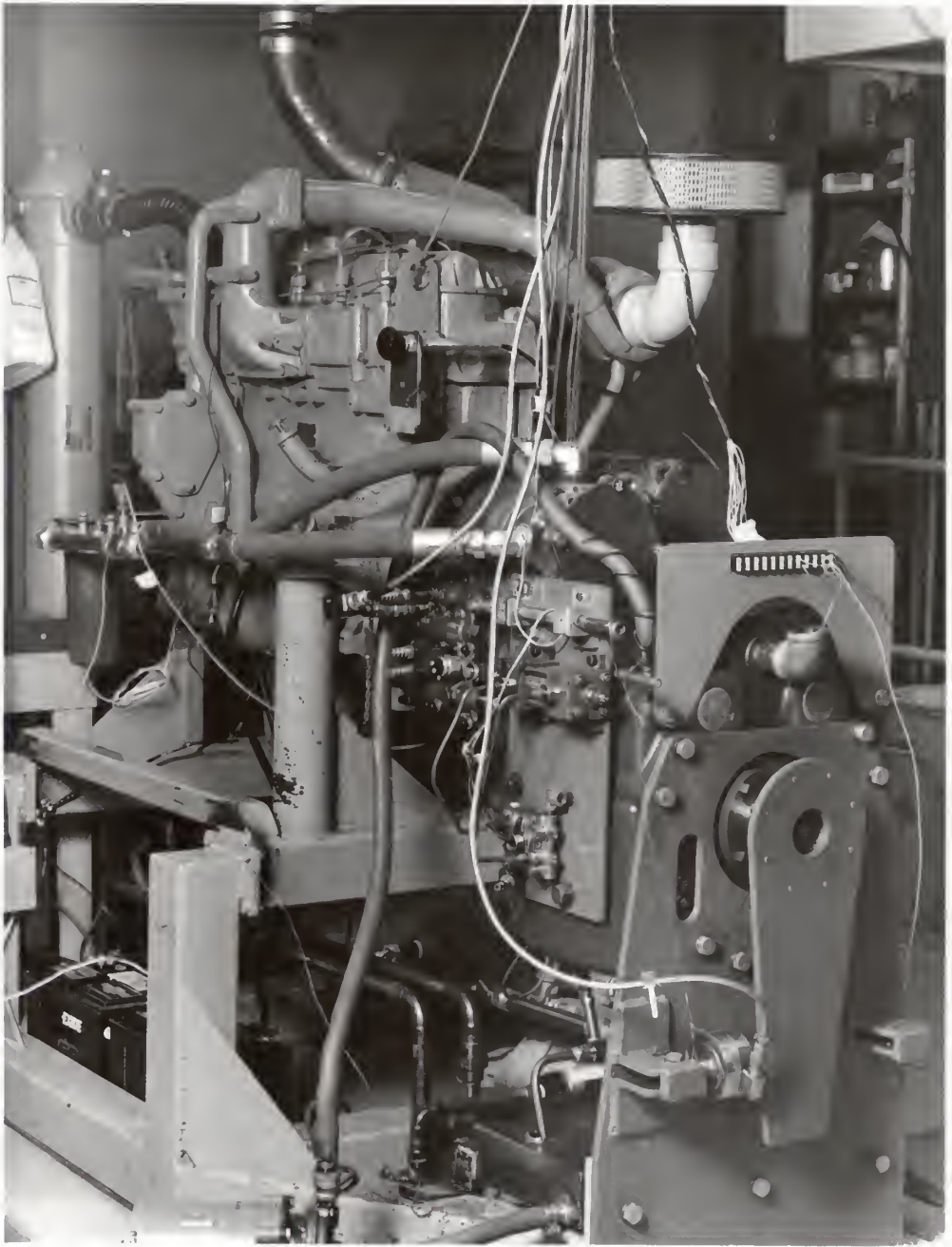


Plate 7. Test Setup From Left Hand Side

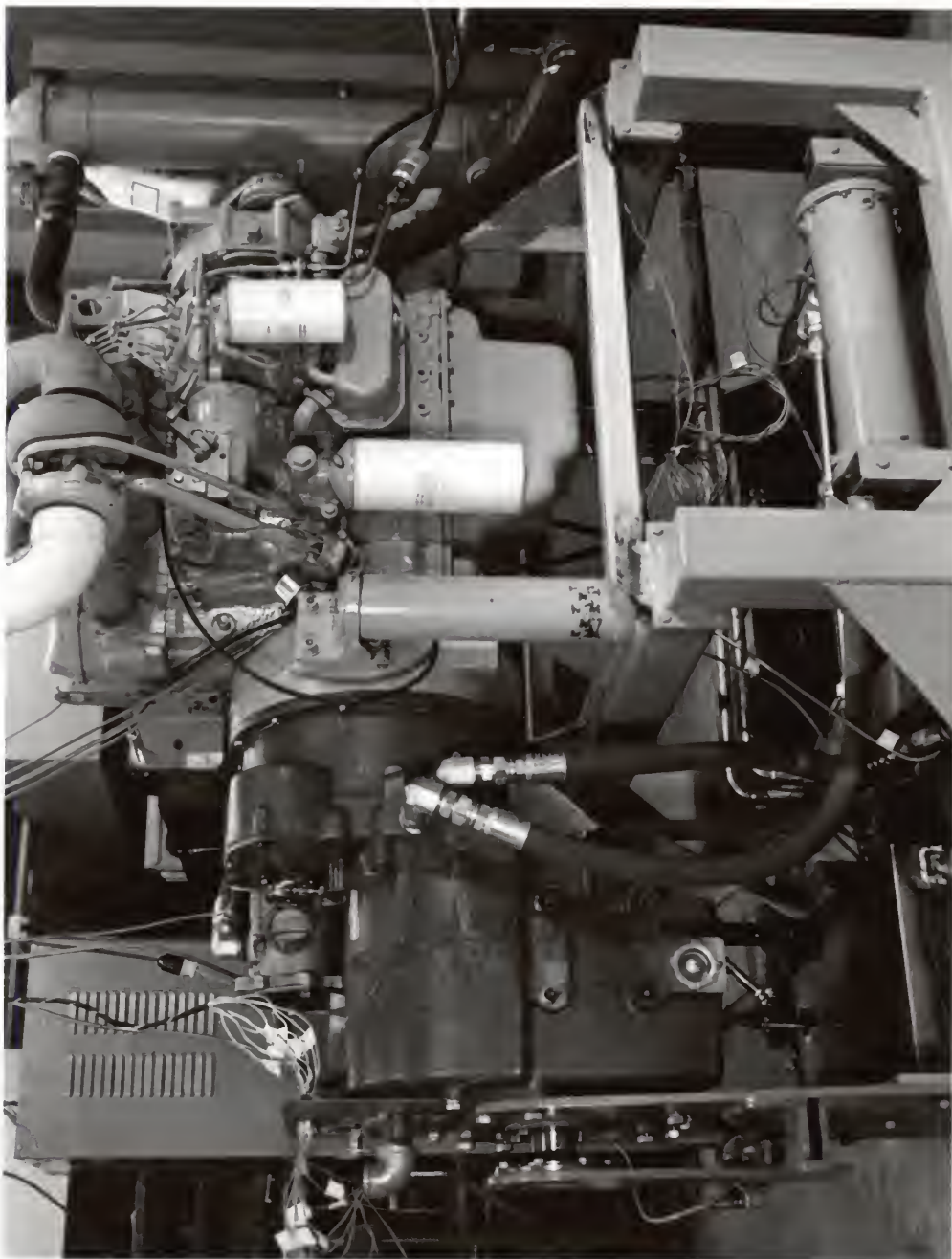


Plate 8. Test Setup From Right Hand Side

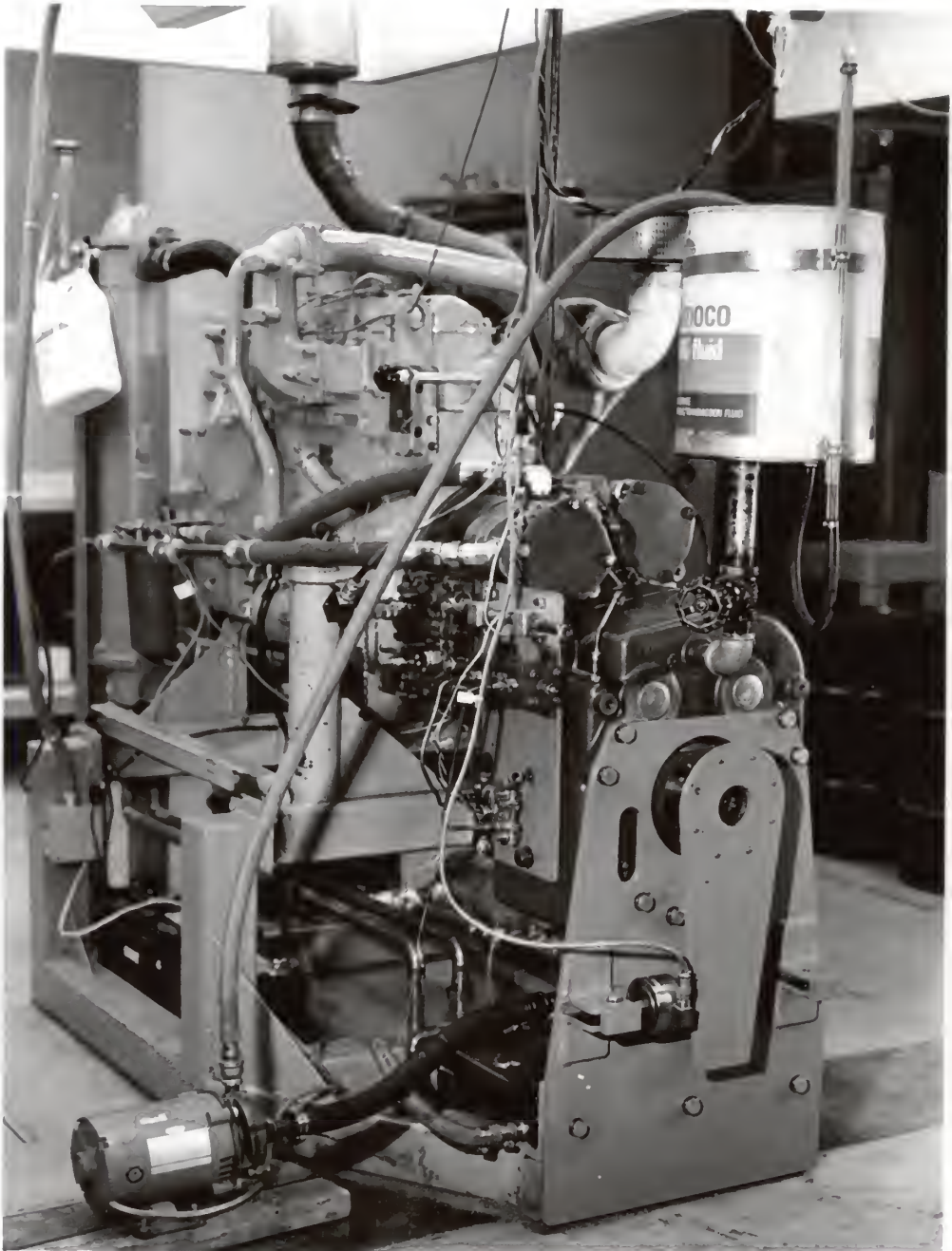


Plate 9. Test Setup for Direct Oil Flow Rate Measurement

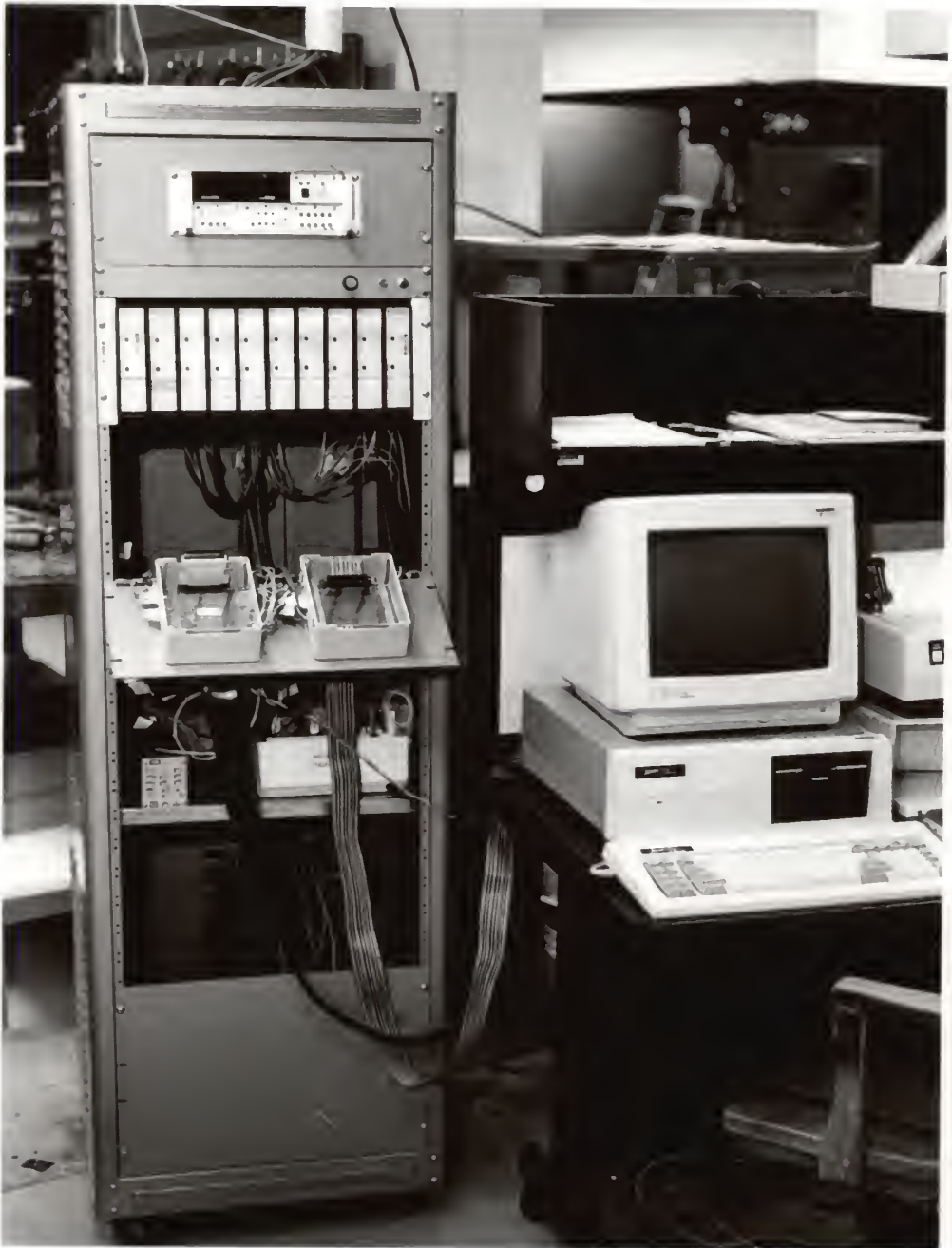


Plate 10. Computer and Instrumentation Cabinet

transmission from neutral into forward.

Clutch Oil Flow Rate

Figure 10a is a summary of the average steady state clutch oil flow rate through the forward directional clutch under slippage at different friction clutch plate speeds. The flow was measured directly by filling the container described previously. The transmission "inching" valve was first preset in order to realize a desired clutch piston "apply" pressure. Then the valve was shifted from neutral into forward causing the directional clutch to slip under constant clutch piston "apply pressure". About 15 seconds were allowed for the clutch to fully engage and the clutch oil flow rate to stabilize. The flow was then measured over a period of 30 - 45 seconds. The temperature of the oil entering the clutch was maintained between 170 F and 180 F. Comparing the difference in oil flow rates between the engaged clutch in Figure 10a and the open (or non-engaged) clutch in Figure 3, it can be noted that for low clutch pressures, the oil flow rates shown in Figure 10a are greater than the oil flow rates shown in Figure 3. The oil flow rate through the clutch actually increases when the clutch is engaged.

Figure 10a, which shows the steady state oil flow rate through a clutch under slippage, can now be combined with

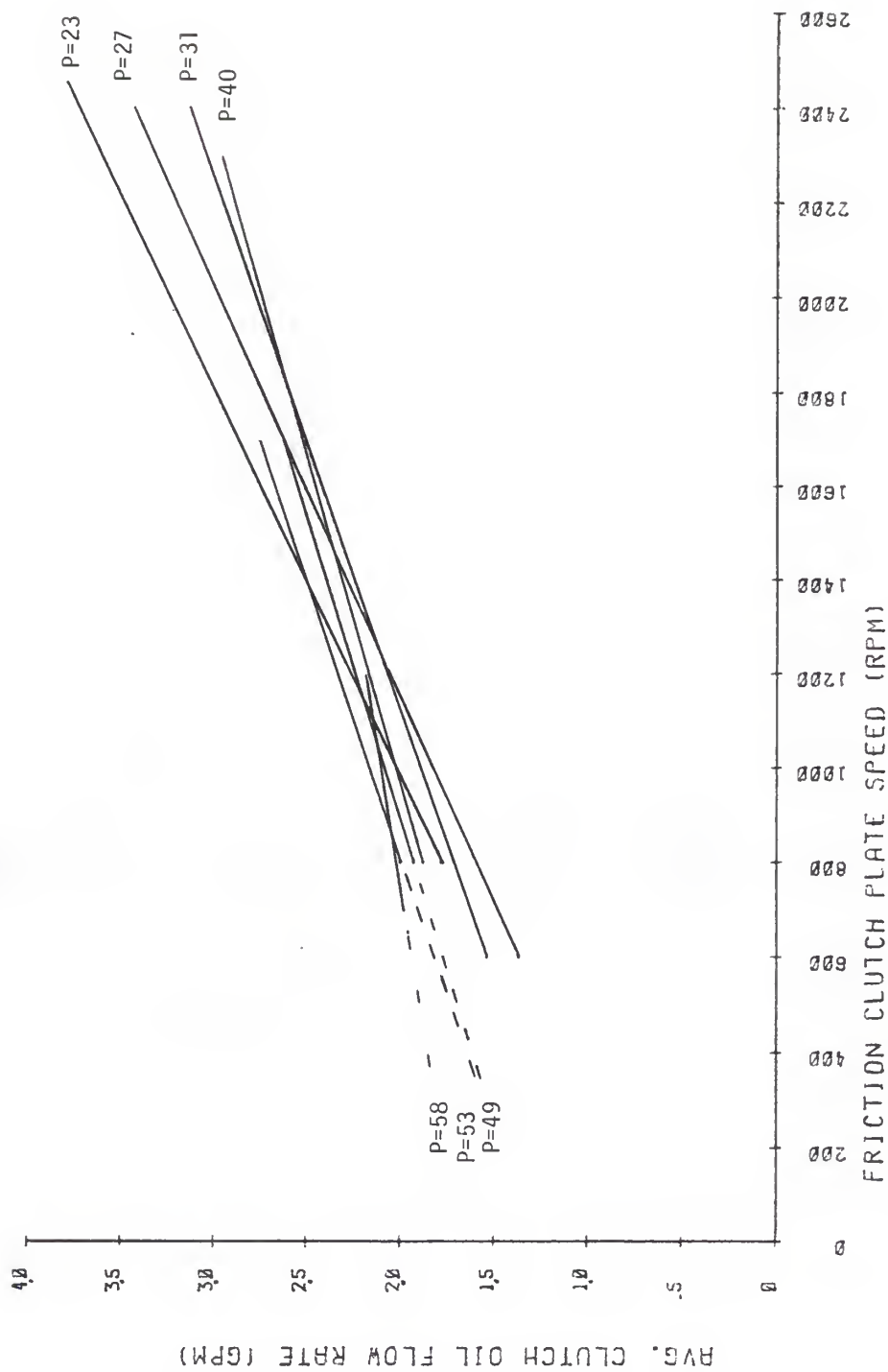


FIGURE 10a. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

Figure 4, which shows the oil flow rate through the friction clutch plate grooves, to determine the oil flow rate over the surface of the reaction clutch plates. Comparison of these two figures show that the oil flow rate through the grooves is typically 15-20% of the total oil flow rate. Therefore, 80-85% of the oil flows over the surface of the clutch plates.

Figure 10a is actually a summary of Figures 10b-10h. Upon inspection, these figures show the individual data points used to obtain Figure 10a. The lines through the data points were obtained by linear regression. The lines showing the average clutch oil flow rates in Figure 10a extend to the maximum clutch plate speed possible for the clutch piston pressures shown. The engine is at full throttle at this point, so no greater friction clutch plate speeds are possible. The clutch oil flow rate initially decreases with increasing pressure behind the clutch piston as shown in Figure 10a. Since the oil pump is driven by the engine, the pump flow is proportional to engine speed. Increasing the clutch piston pressure causes the torque converter speed ratio to decrease. Therefore to maintain a constant friction clutch plate speed under increasing clutch piston pressure, the engine speed must increase. Thus in Figure 10a the change of slope with increasing clutch piston pressure is due to the change in pump oil flow rate.

The maximum oil flow rate through an open directional

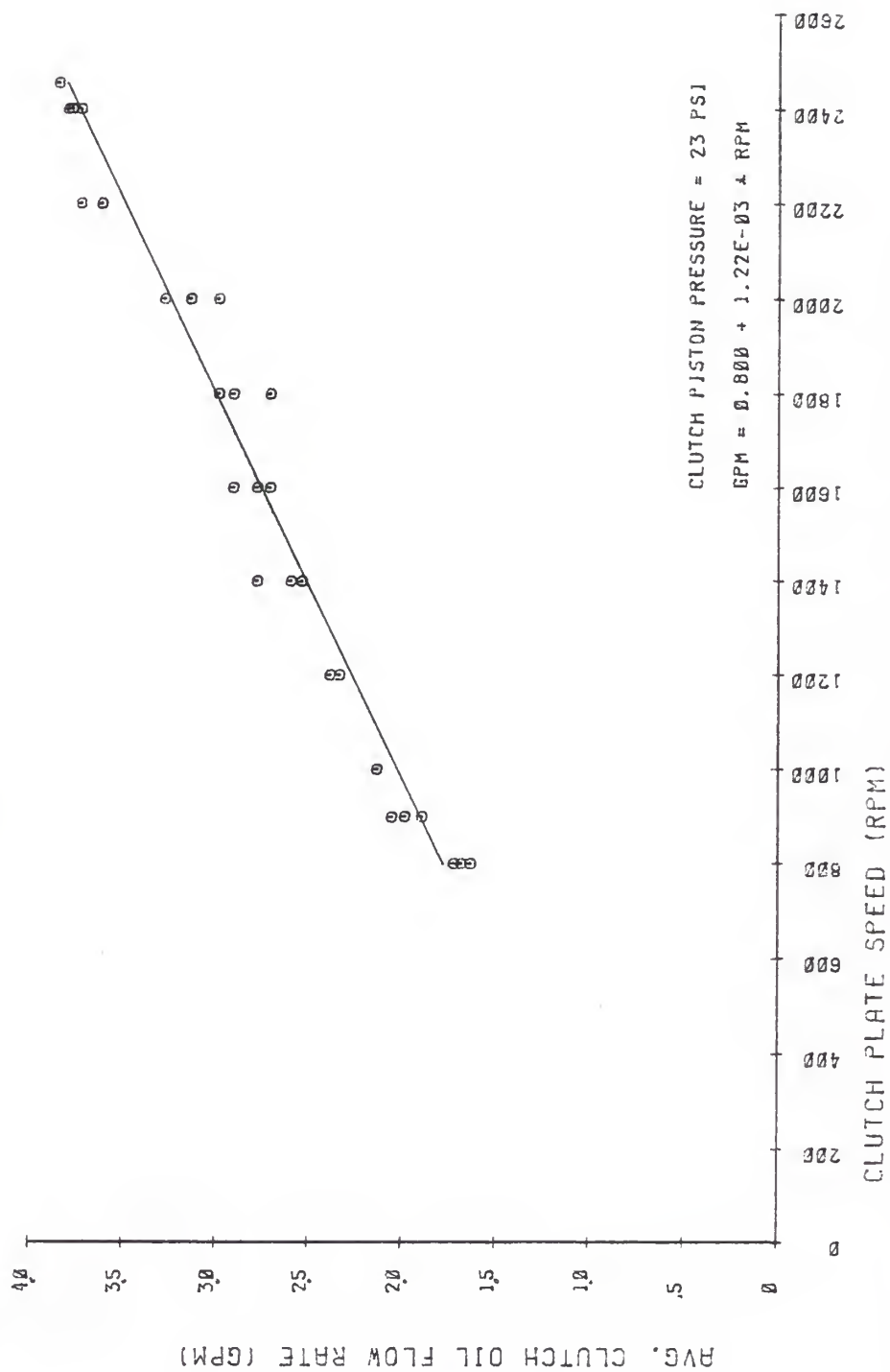


FIGURE 106. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

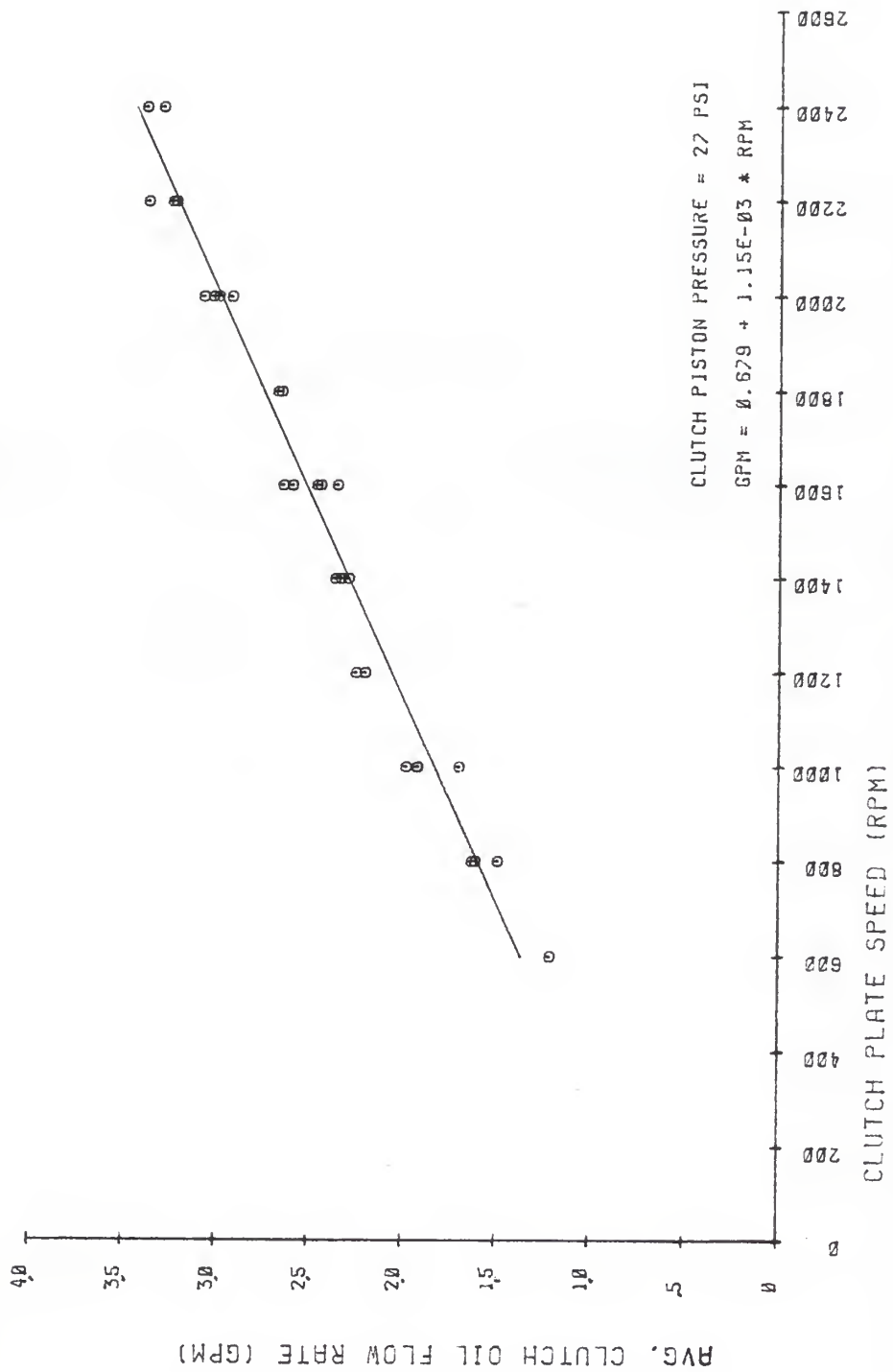


FIGURE 10c. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

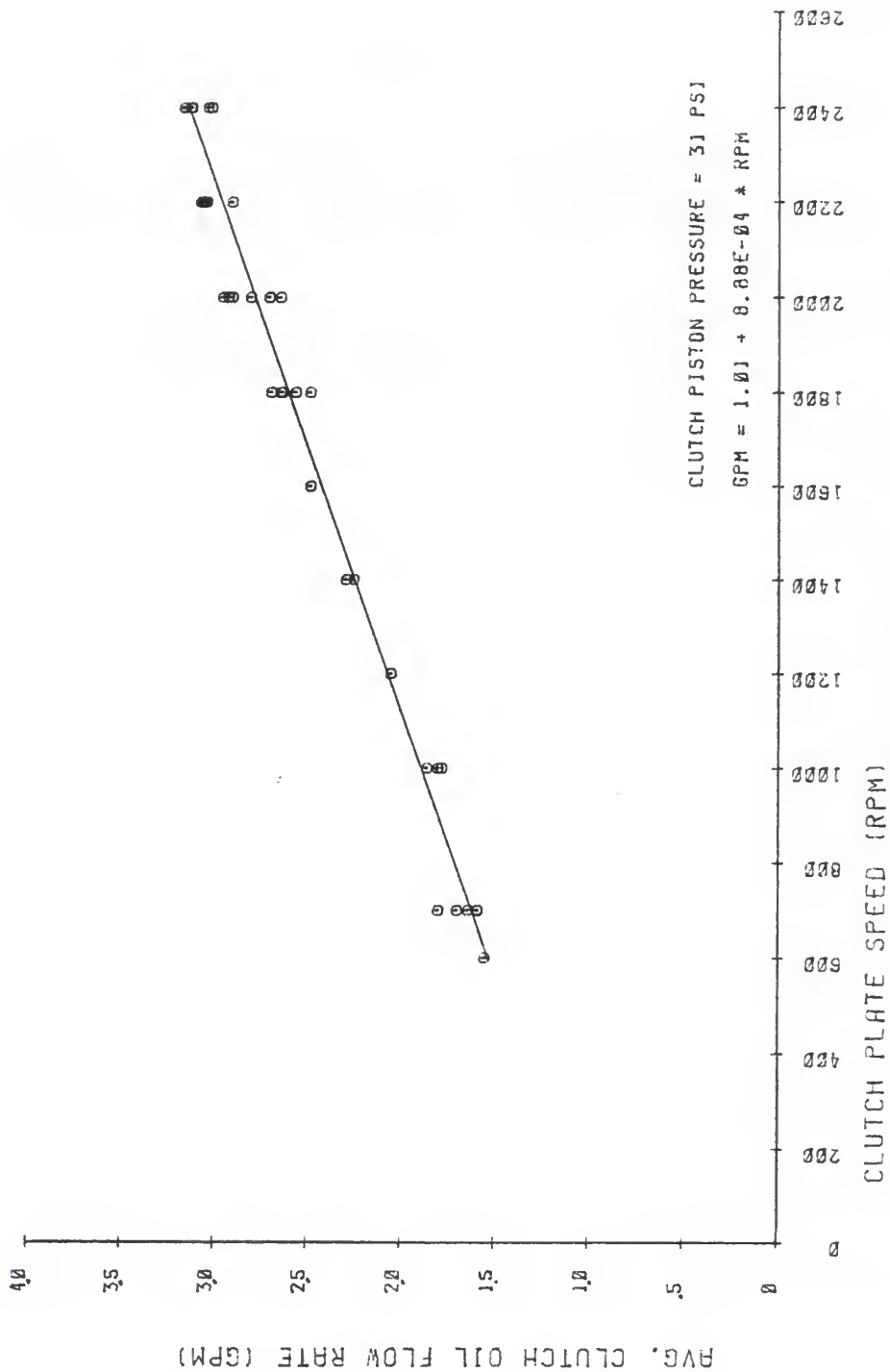


FIGURE 10d. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

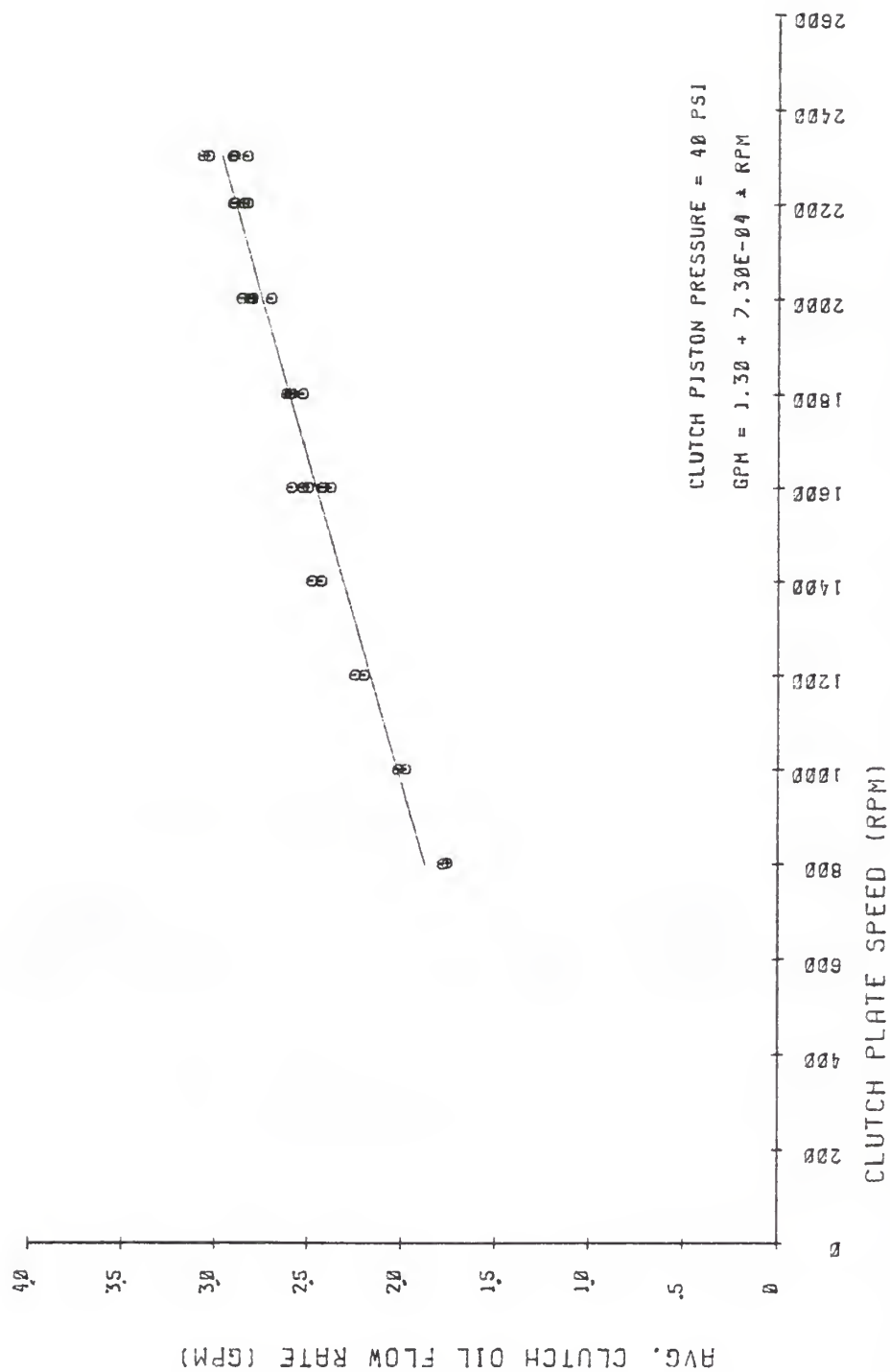


FIGURE 10e. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

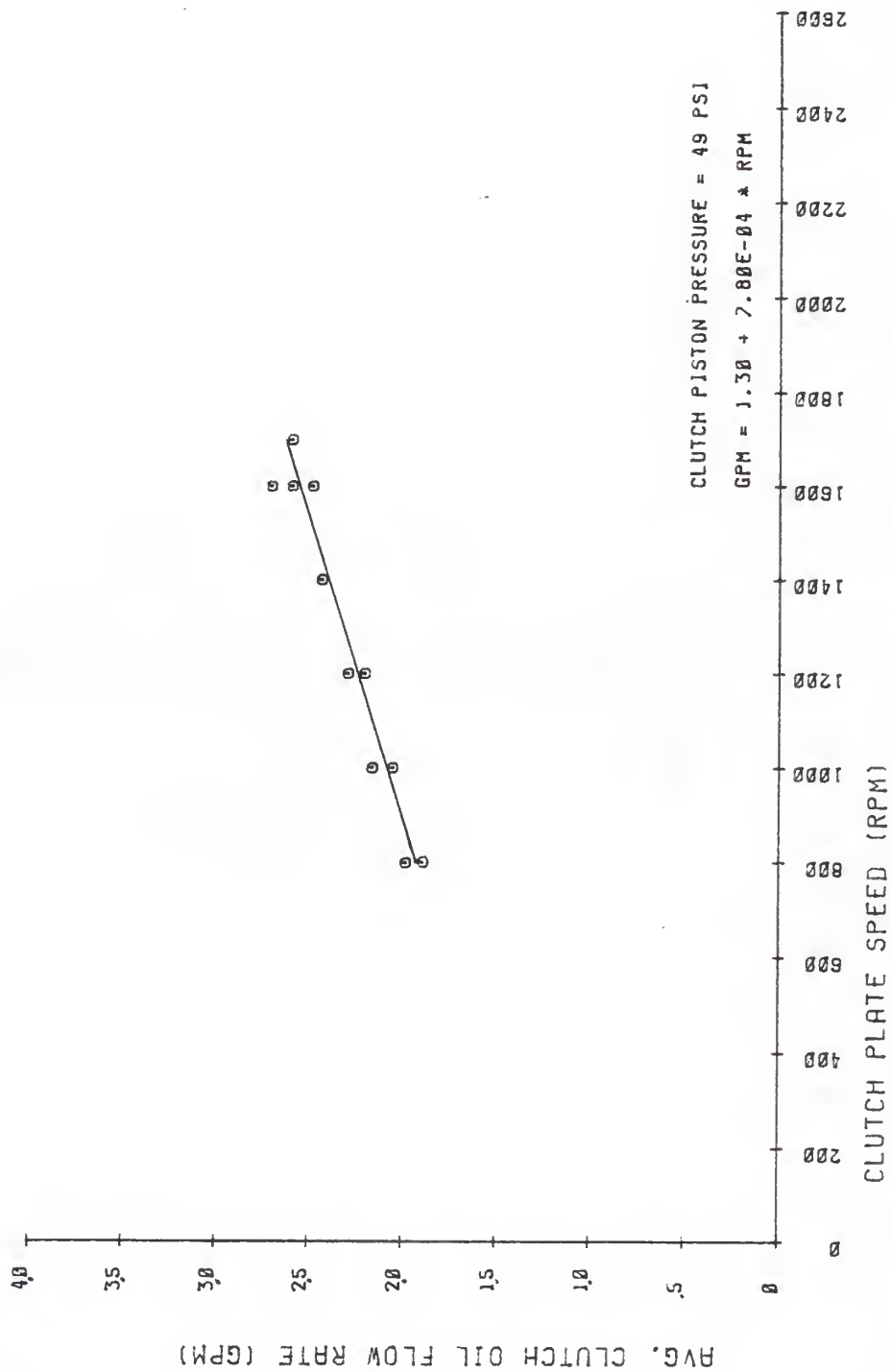


FIGURE 104. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

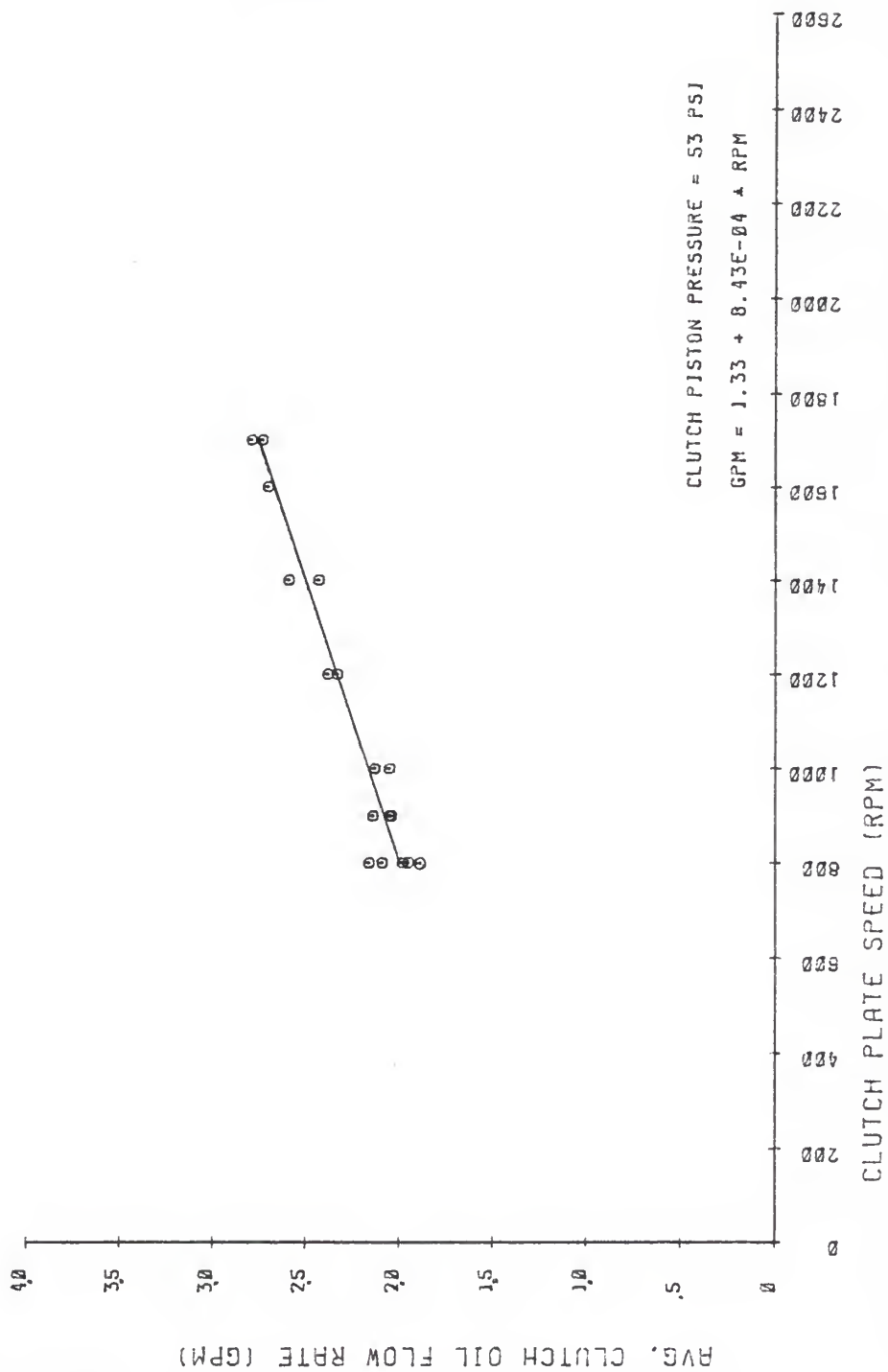


FIGURE 109. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

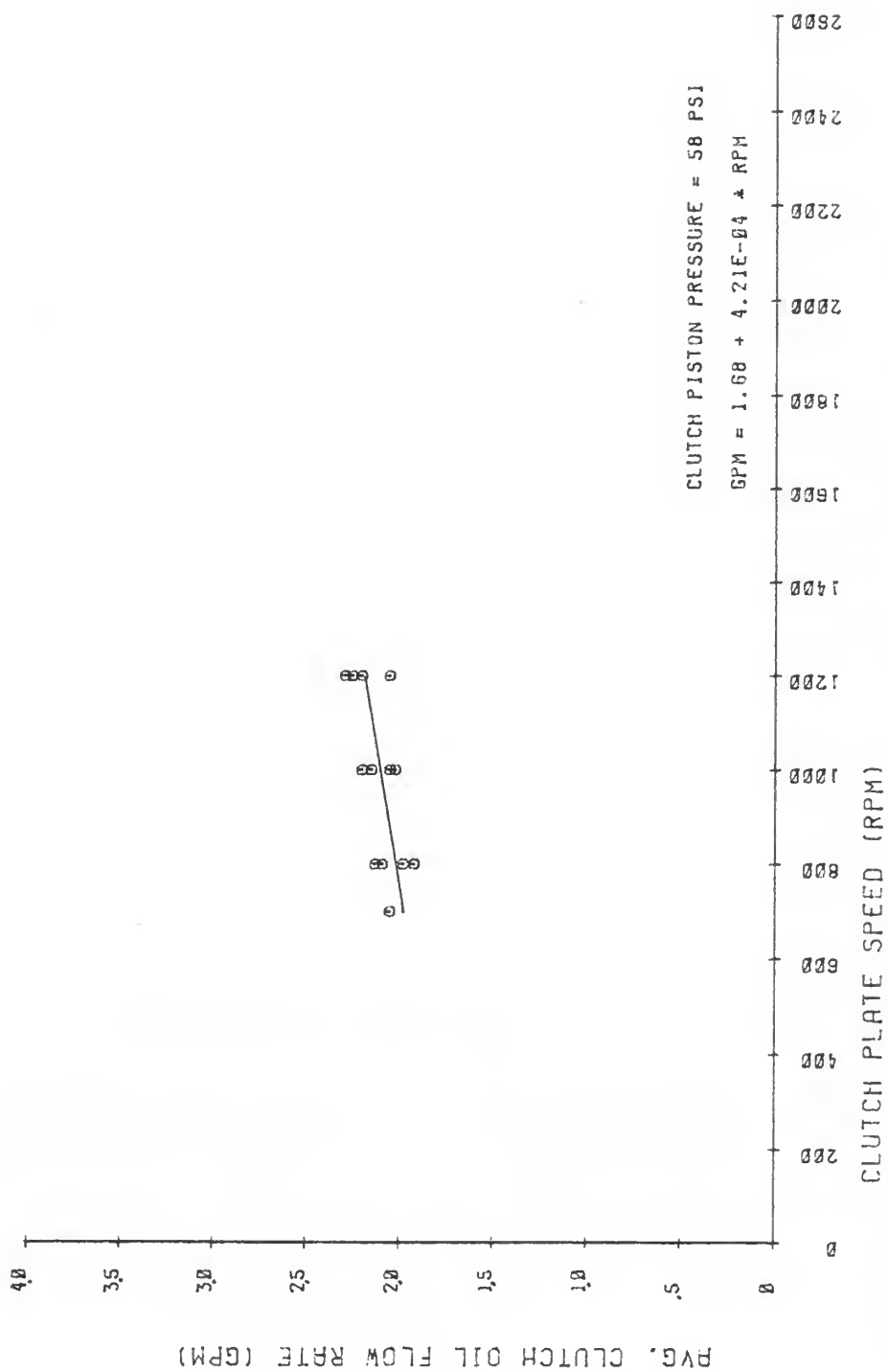


FIGURE 10b. AVERAGE STEADY STATE OIL FLOW RATE (ENGAGED CLUTCH)

clutch (previously shown by Figure 3) is 3.4 gpm at a friction clutch plate speed of 2400 rpm. The oil flow through an engaged clutch with 23 psi behind the clutch piston results in over 3.7 gpm (see Figure 10a) at a clutch plate speed of 2400 rpm. It was noticed that a small pressure behind the clutch piston actually increased the flow through the clutch. However, increasing the clutch piston pressure beyond a certain point reduced the flow as expected. A possible explanation for this behavior is that the open clutch restricts flow by causing the clutch plates to flutter or vibrate, thus causing turbulent flow regimes. Engaging the clutch by pressurizing the clutch piston slightly then causes the flow over the plate surface to become laminar. However, the pressure behind the piston is not yet great enough to restrict the flow by clamping the clutch pack very tightly.

Figures 11a - 11e show the oil flow rate, as measured by the turbine flowmeter, through the forward directional clutch as a function of time for a step input of heat. The clutch is slipped at a constant friction clutch plate speed and a constant clutch piston pressure during the slip time. Note that the oil flow rate initially increases to a maximum value during the time period when the clutch pack is closing. Results from the steady state oil flow rate tests (see Figure 10a where some of the oil flow rates are greater than those in Figure 3) indicate that this data is correct,

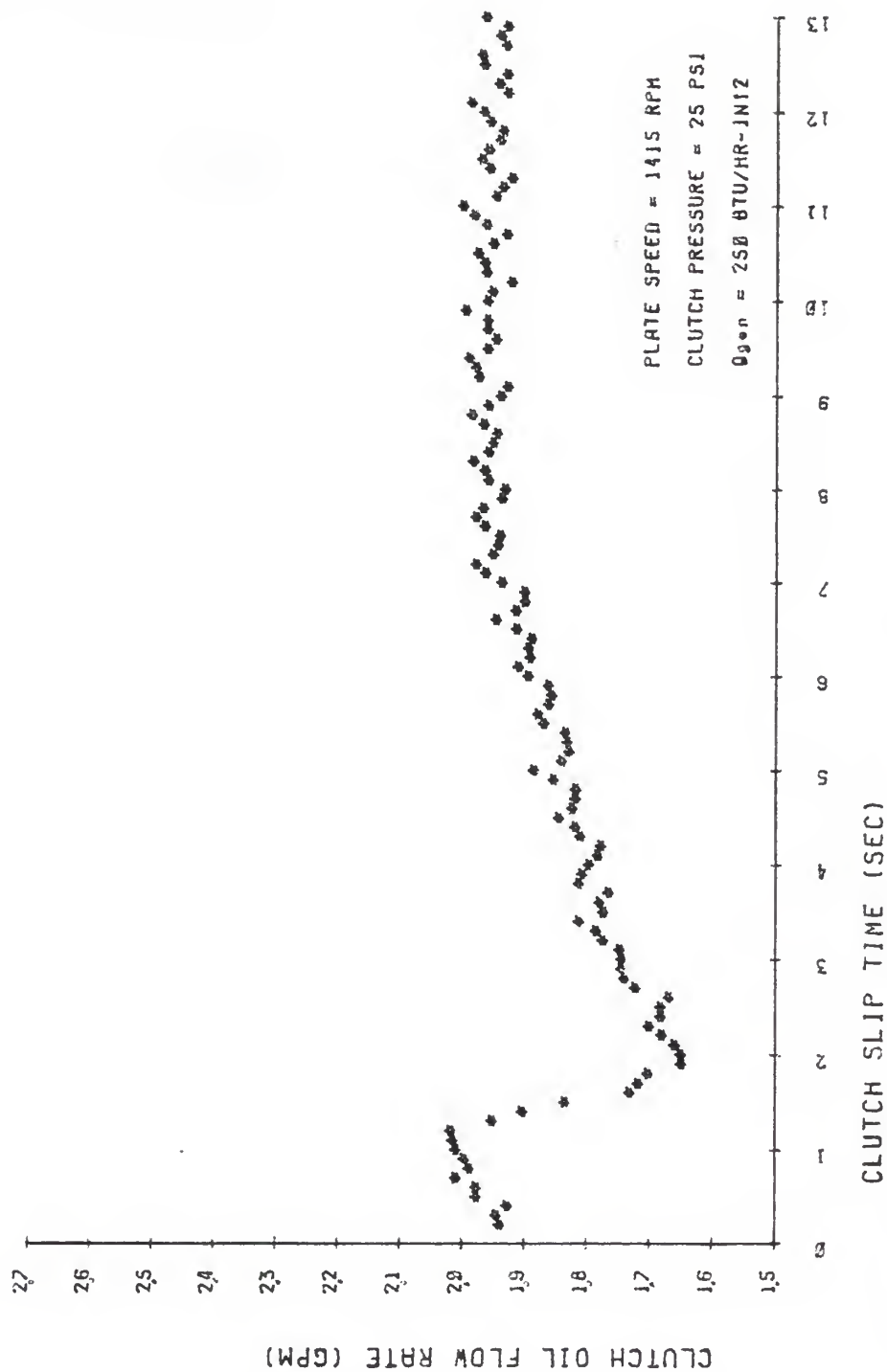


FIGURE 11a. OIL FLOW RATE THROUGH ENGAGED CLUTCH

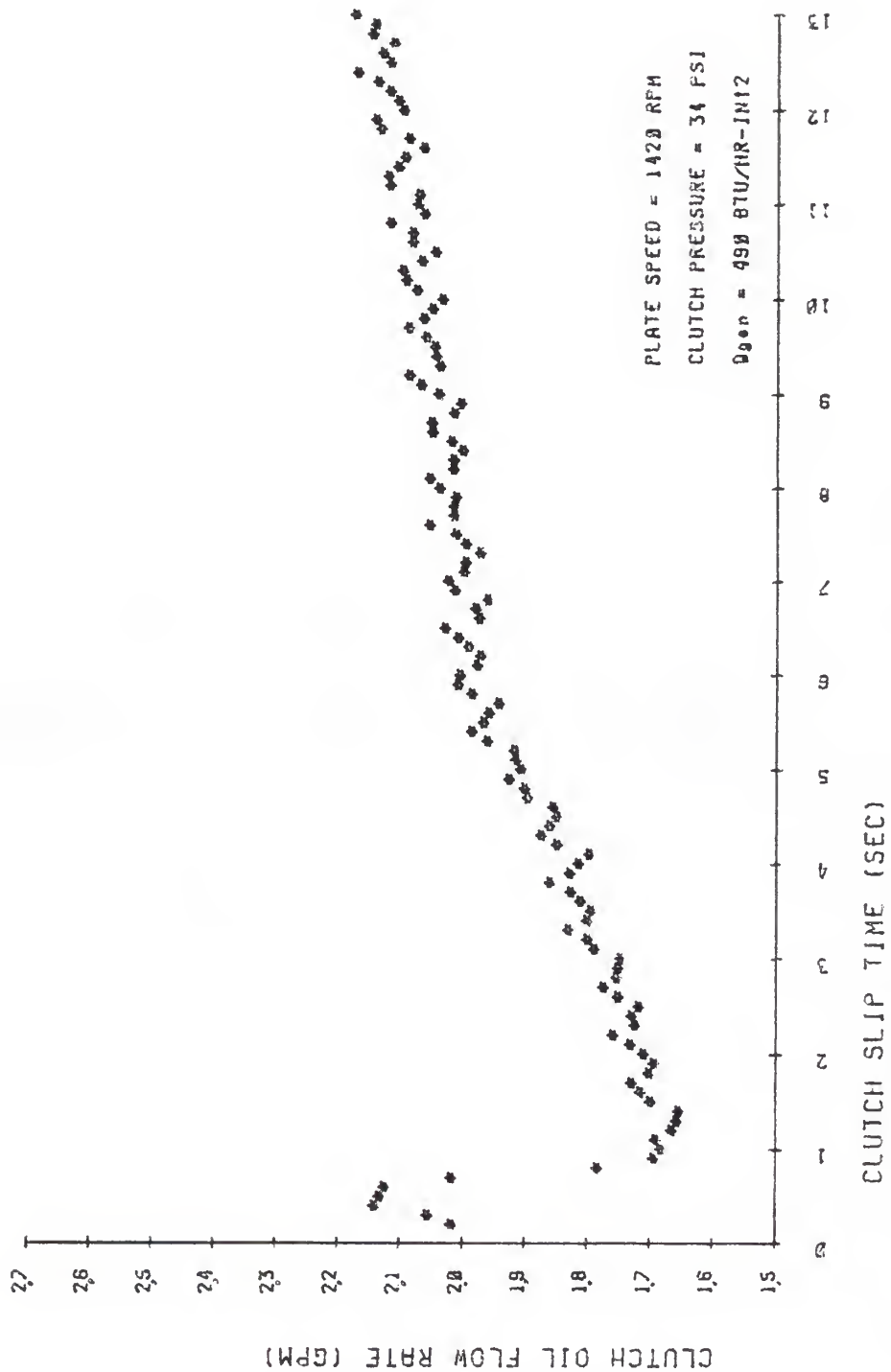
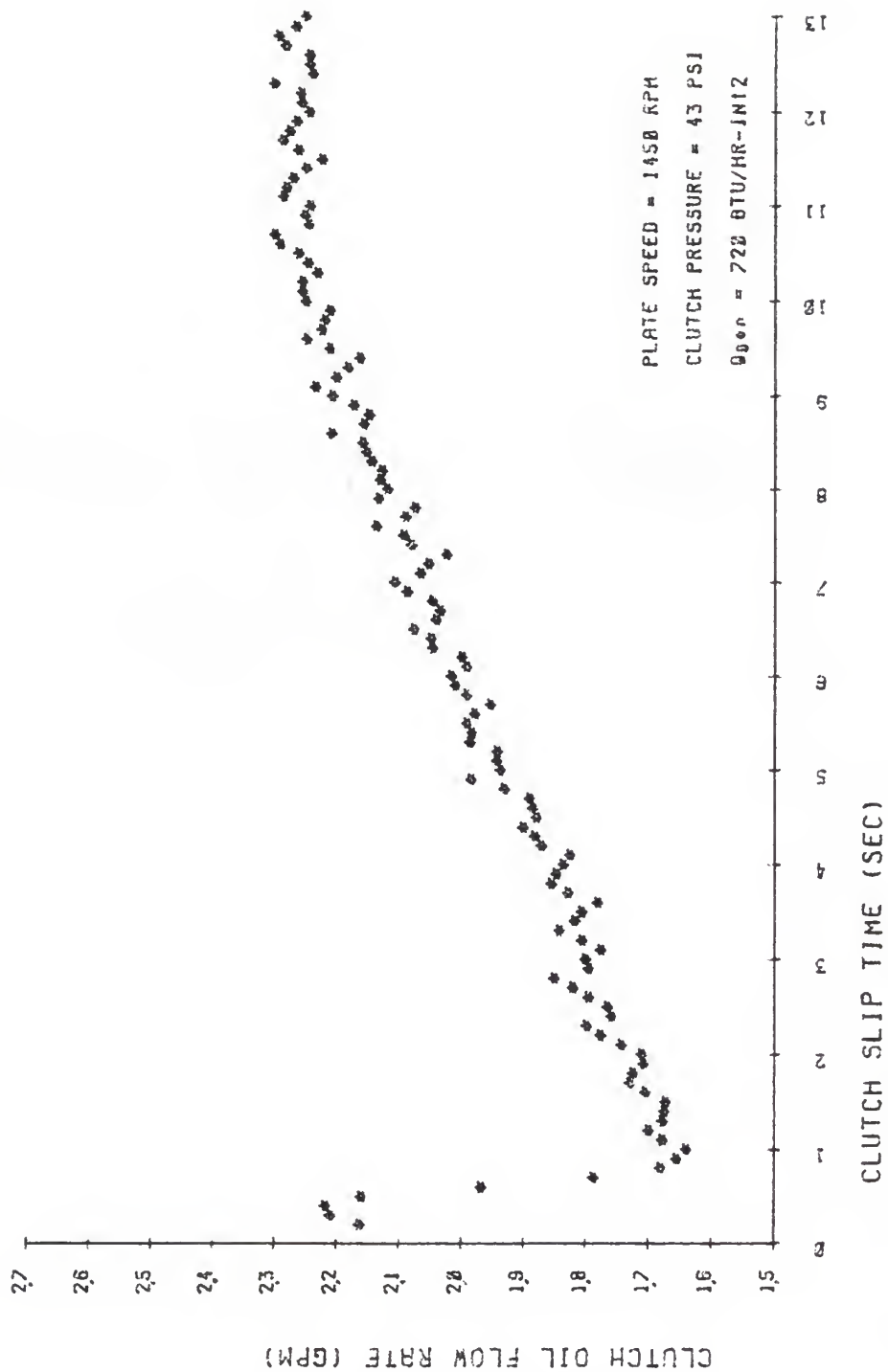


FIGURE 11b. OIL FLOW RATE THROUGH ENGAGED CLUTCH



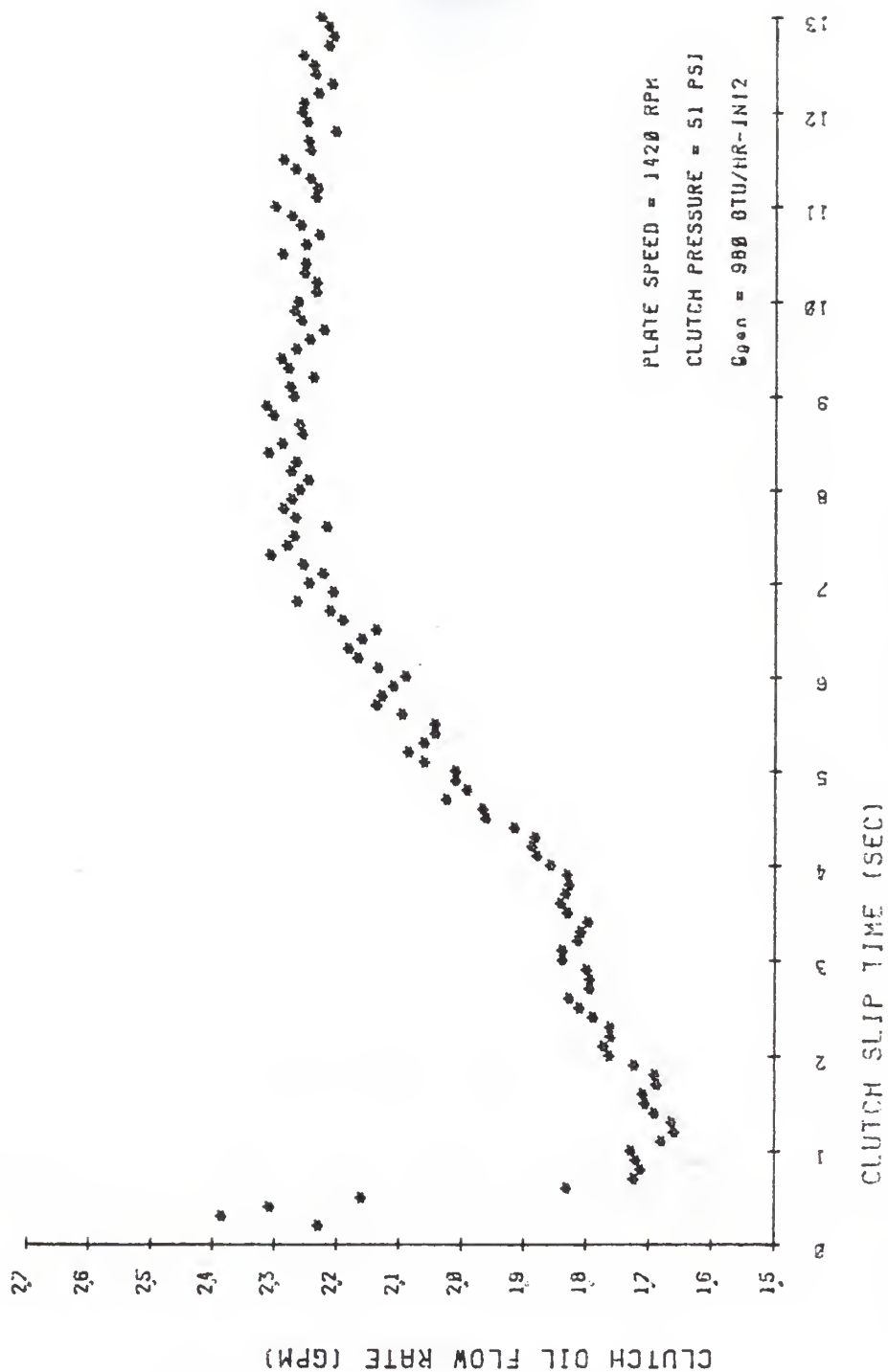
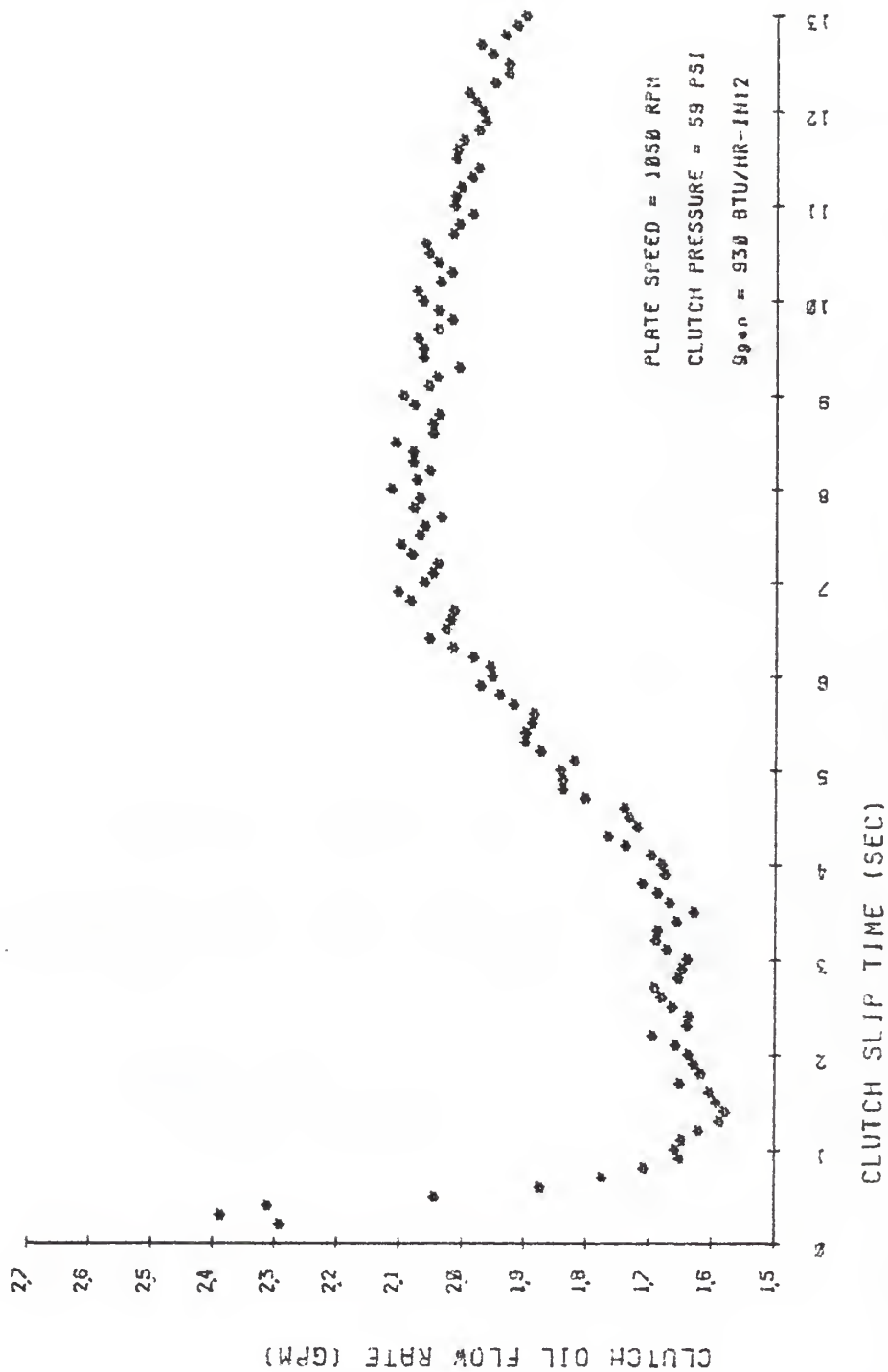


FIGURE 11d. OIL FLOW RATE THROUGH ENGAGED CLUTCH



and not caused by an error in the flowmeter response of the test setup. The flowmeter itself is not in error, but rather the time required for the flowmeter to detect a transient within the shroud may cause a response error. The flowmeter is about two feet down stream from the shroud. The fluid boundary layer prevents physical contact of the plate surfaces during this time, and even seems to augment the flow. As mentioned above, one possible explanation for this change in flow may be a transition from turbulent to laminar flow across the plate surfaces. Figure 11a shows this phenomena in the greatest detail because the clutch engaged more slowly here, because of lower clutch piston pressure than in Figures 11b - 11e.

Another possible explanation is that this momentary flow rate increase, occurring when the clutch piston is engaged, is the result of excess oil being squeezed from between the clutch plates and from within the cavity inside the clutch plates, thereby reducing the volume within the clutch cylinder (see Figure 1). The sealed bearing prevents any oil flow path out of the clutch except through the clutch plates. With only about 0.100 inches of total clutch plate stackup clearance in the clutch (or about 0.005 inches between each clutch plate), the decrease in clutch volume occurring when the clutch is engaged is insignificant (consider the flow increase versus clutch engagement time involved in Figure 11a).

At some finite clutch pressure, as shown in Figure 11a (also shown in Figures 11b- 11e), the oil flow rate rapidly decreases to a minimum value. The clutch piston pressure at which this oil flow rate decrease occurs is not known. However, the clutch piston pressure has increased to such an extent that the reaction clutch plate and friction clutch plate surfaces finally do contact, resulting in frictional heating of the reaction clutch plate. Figures 12a - 12e show the oil exit temperature and oil flow rate as functions of time. Comparing the oil exit temperature against the oil flow rate determines the point in time at which heat was input into the reaction clutch plates. The oil flow rate then increases, and in Figures 11d - 11e, the flow rate reaches another local maximum value, after which the flow rate decreases. This flow increase is due to factors which shall be discussed in the paragraphs which follow.

The oil flowing through the clutch turbine flowmeter, located downstream from the clutch, contains a considerable amount of entrained air, even when the clutch is disengaged. The air, caused by turbulence within the clutch, is mixed into the oil as the oil flows between the clutch plates. This air causes the turbine flowmeter to read high. An effort was made to estimate this error by filling a beaker with oil that had exited the flowmeter. At low engine speeds with the transmission in neutral, the oil contained about 2% air. This percentage was determined by allowing

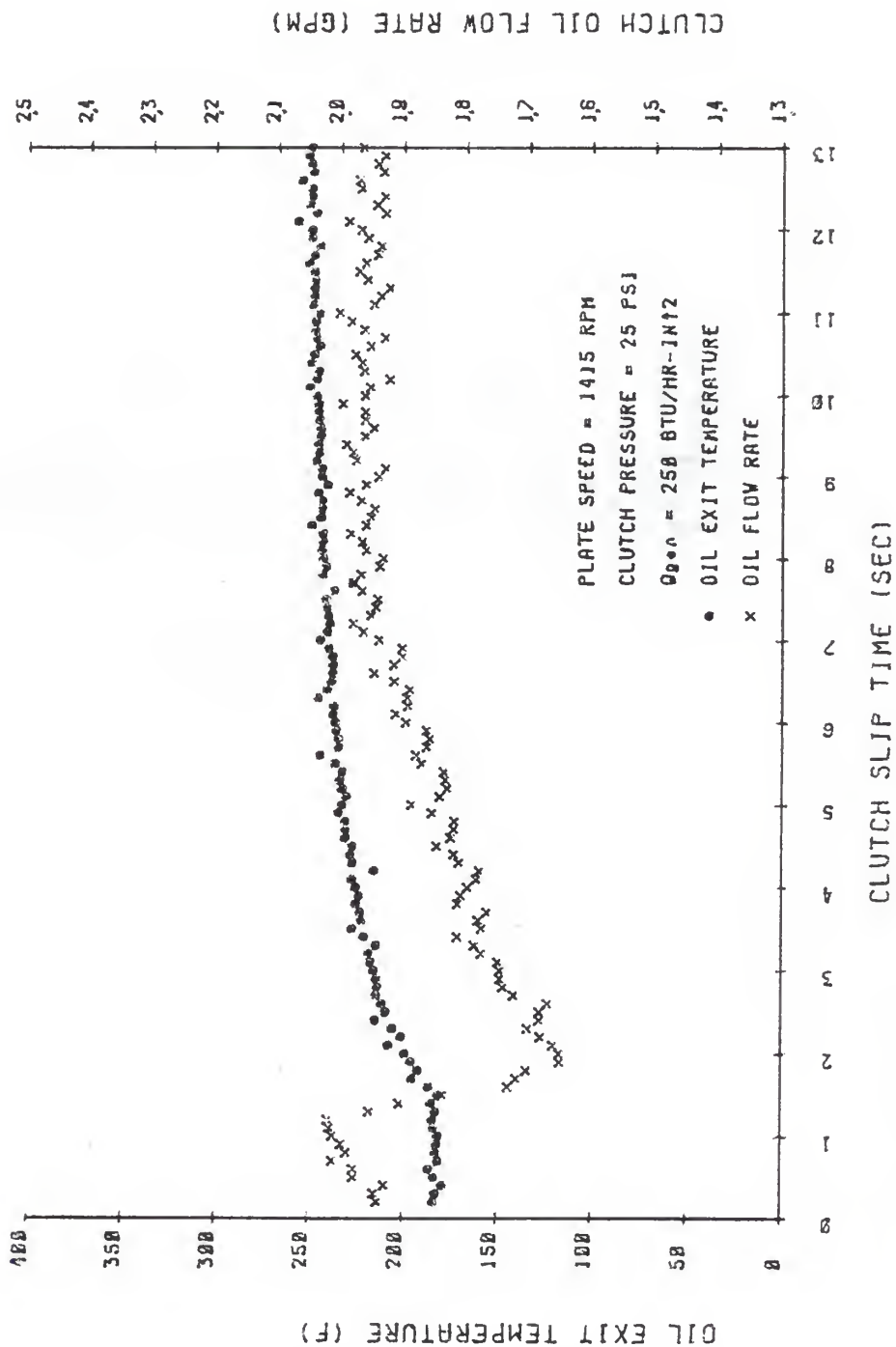


FIGURE 12a. OIL FLOW RATE & OIL EXIT TEMPERATURE

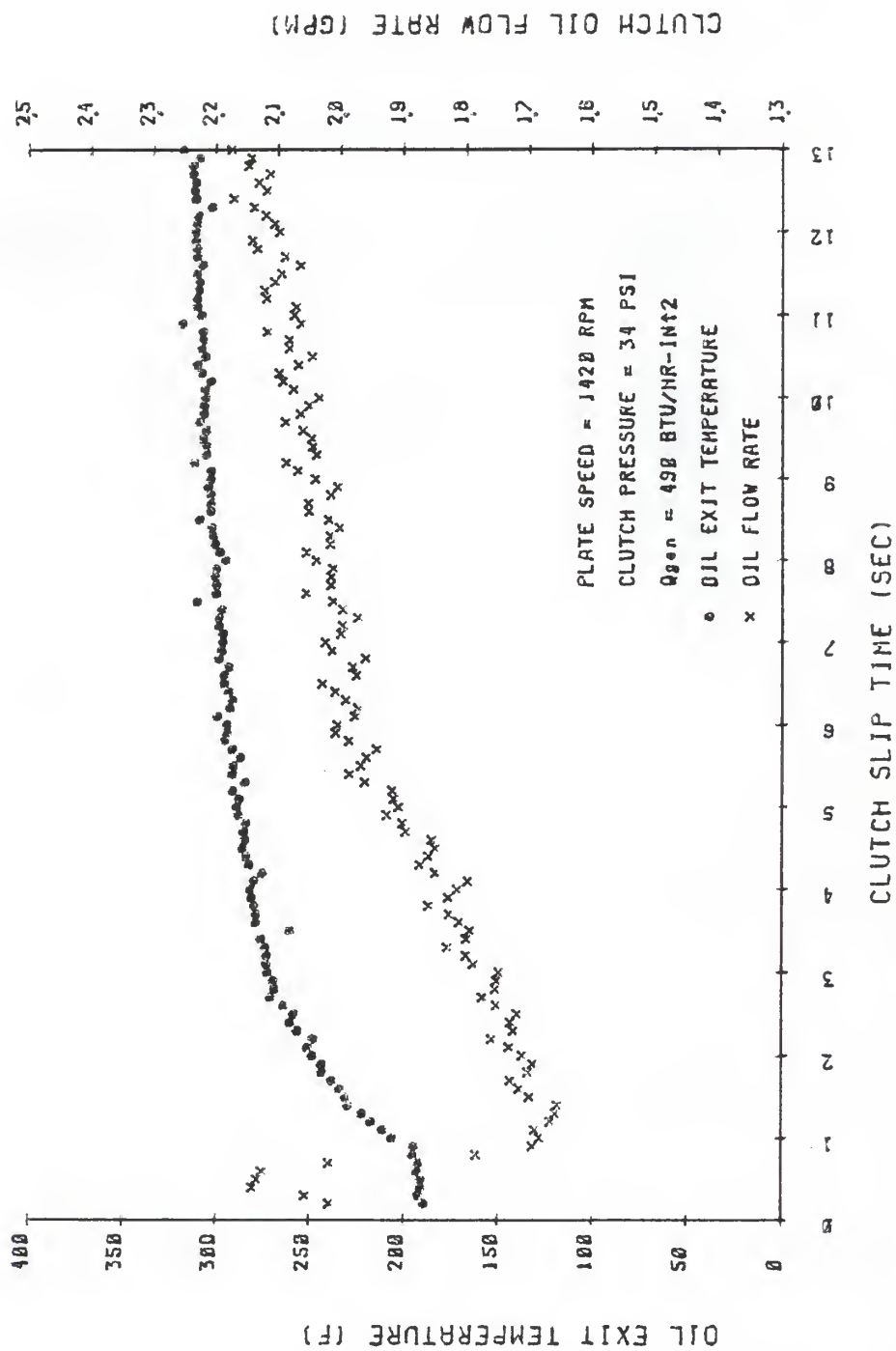


FIGURE 12b. OIL FLOW RATE & OIL EXIT TEMPERATURE

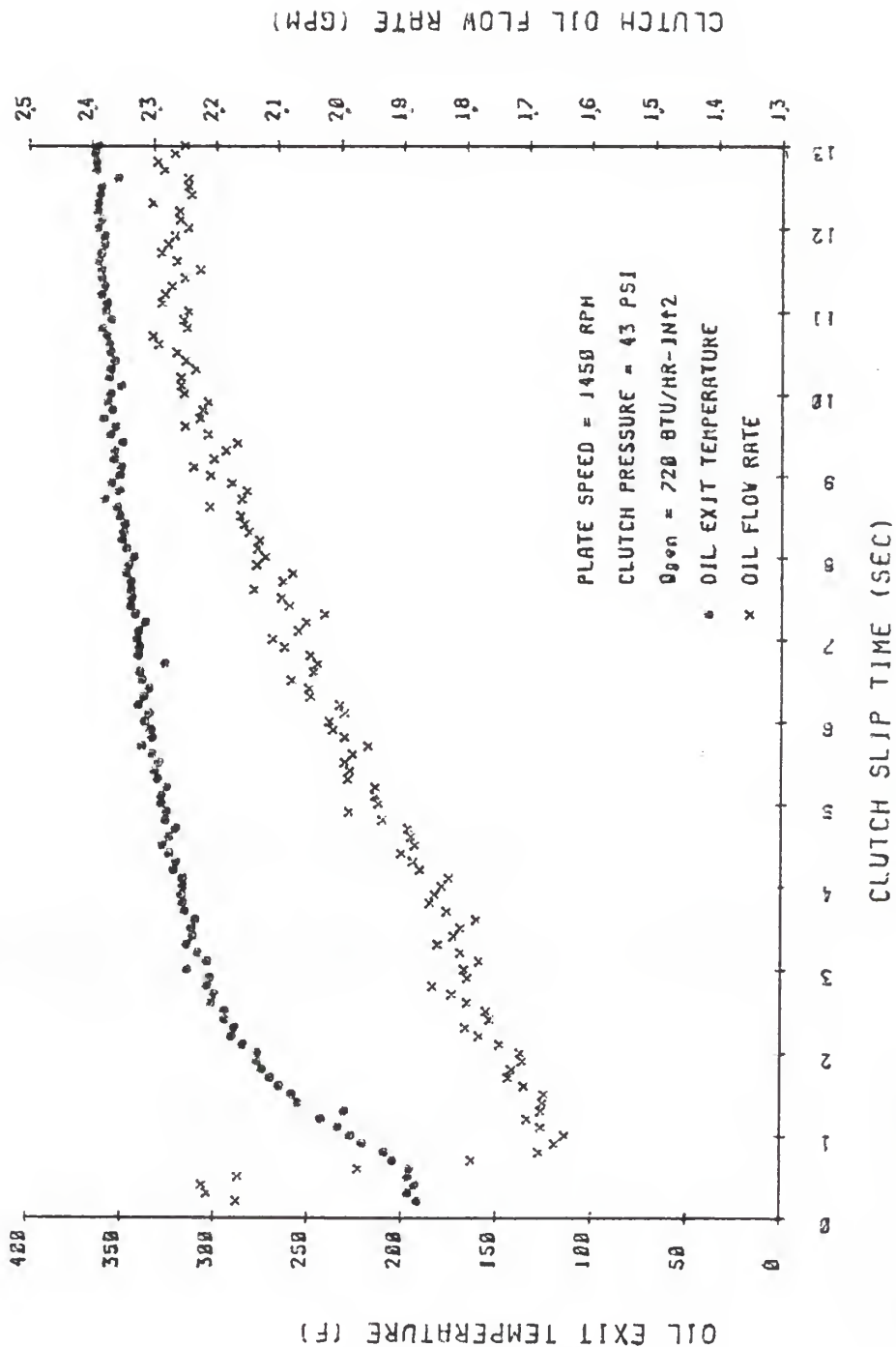


FIGURE 12c. OIL FLOW RATE & OIL EXIT TEMPERATURE

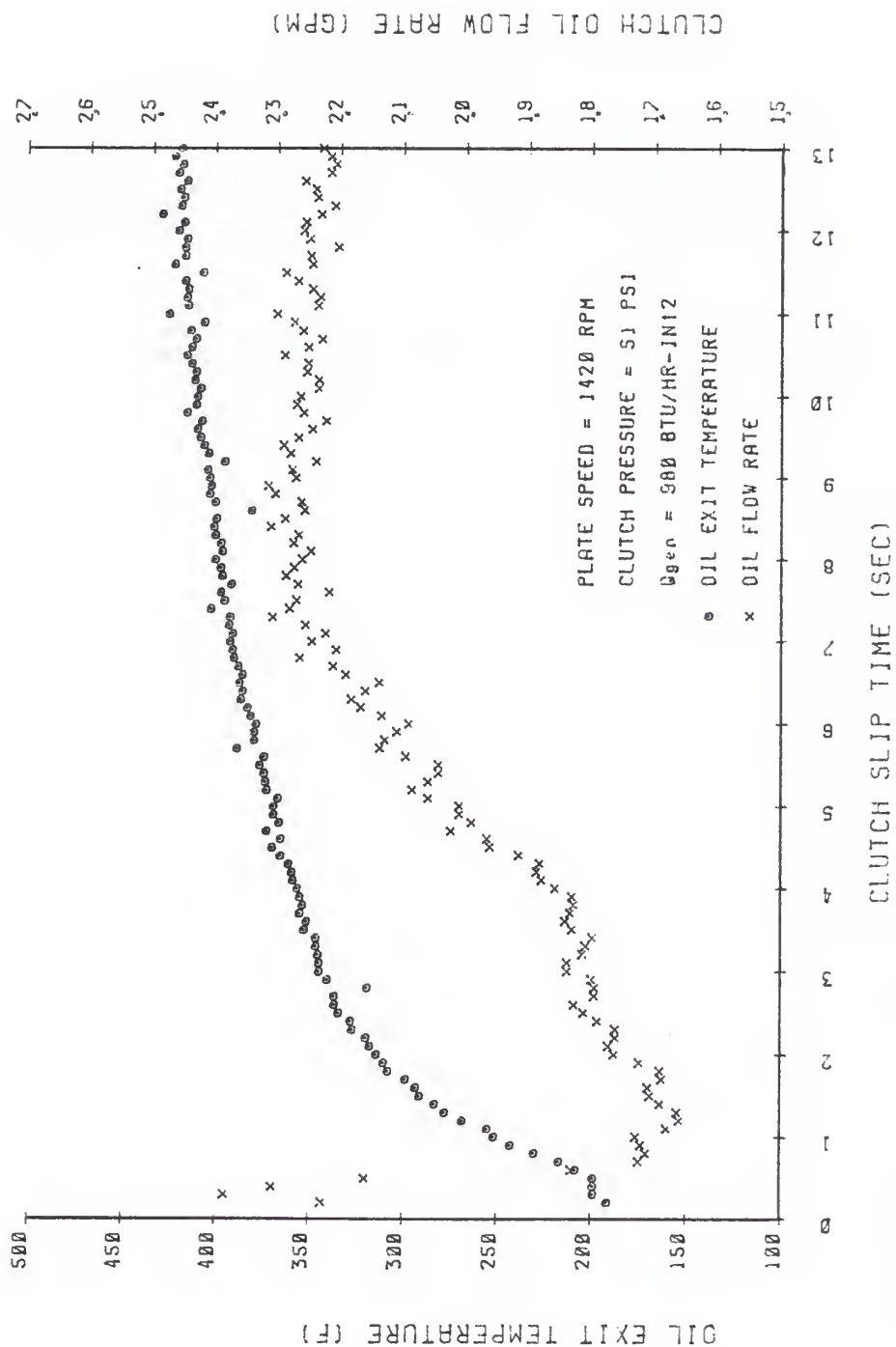


FIGURE 12d. OIL FLOW RATE & OIL EXIT TEMPERATURE

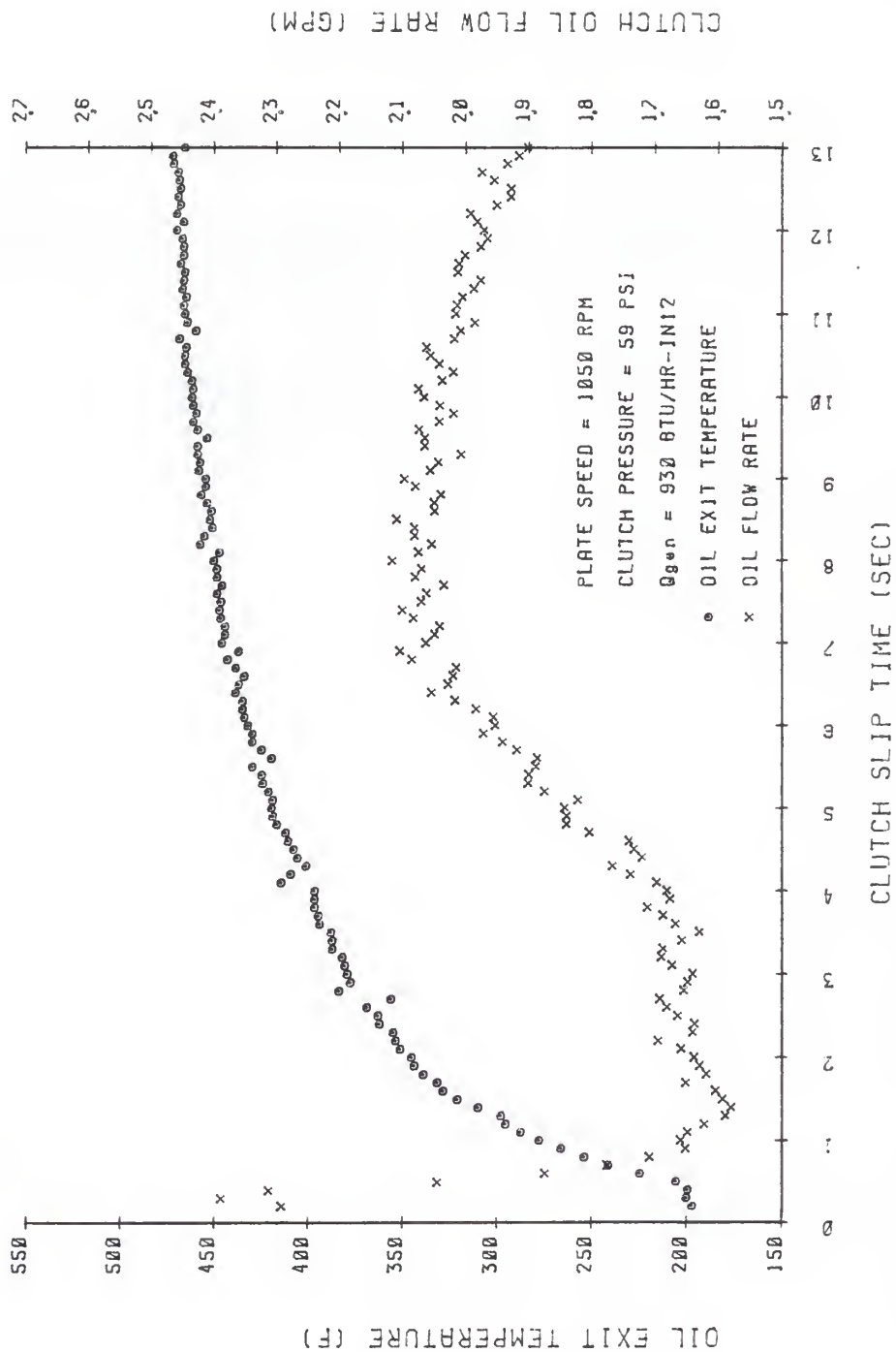


FIGURE 12e. OIL FLOW RATE & OIL EXIT TEMPERATURE

the air bubbles to settle out of the oil and observing the resulting decrease of oil level within the beaker. Increasing the engine speed to high idle resulted in approximately 3% air. Lightly engaging the clutch caused the plates to be squeezed together and the plate temperature to increase. Foam began to appear on the surface of the oil within the beaker but the quantity of foam was difficult to measure because it dissipated rapidly. The oil at this point contained at least 5% air, and even some oil vapor. Tests documented in Reference 3 indicate that up to 14% air may be dissolved in a typical transmission fluid. Therefore, a "solid" layer of oil does not exist between the friction clutch plates and reaction clutch plates, but rather the space is filled with a mixture of oil and air [3].

During the tests which were conducted it was determined that the measured reaction clutch plate temperature can exceed 500 F when the clutch is slipped over a period of several seconds. When this is the case, a considerable amount of vapor appears in the oil stream. The oil is heated as it flows over the reaction clutch plate surfaces and also through the grooves in the friction clutch plates. Since the oil gives off increasing amounts of vapor with increasing temperature, two-phase flow can become significant at these higher clutch plate temperatures. And, since the turbine flowmeter used was calibrated for liquid only, it is likely that two-phase flow accounts for some of

the increase in oil flow rate shown in Figures 11a - 11e and Figures 12a - 12e.

The oil flow rate shown in Figure 12e (also shown in Figures 12a - 12d) increases to a local maximum point, after which the oil flow rate decreases. At this local maximum point, the oil/air/vapor mixture is stabilized in quality, so the oil flow rate as measured by the turbine flowmeter no longer increases. This maximum flow point does not seem to occur at any specific reaction clutch plate temperature. After the maximum flow point is reached, the oil flow rate then decreases, as shown in the last portion of the curve in Figure 12e.

In summary, it is believed that the true oil flow rate does not increase with time, but rather is either constant or else decreases with increasing reaction clutch plate temperature. The error in oil flow rate measurement could be significantly reduced by measuring the oil flow rate upstream from the clutch. Relocation of the flowmeter to allow this is possible. However, extensive modification of the test setup will be required. Accurate oil flow rate measurement is critical since the oil flow rate directly affects the heat transfer coefficient calculation in Equation 4.

Reaction Clutch Plate Temperatures

The temperature of the oil exiting the clutch is plotted with the reaction clutch plate centerline temperature in Figures 13a - 13e for a step heat input. The plate centerline temperature lags the oil exit temperature during the first two seconds of clutch engagement. This is because the heat into the reaction clutch plate is first generated at the plate surface (where the oil exit temperature thermocouple is located) and then conducted to the center of the plate (where the reaction clutch plate thermocouple is located). Initially, during this step heat input, the plate surface temperature will be greater than the plate centerline temperature because of the thermal energy storage due to the mass of the plate. Since the temperature of the oil exiting the clutch is dependant upon the reaction clutch plate surface temperature, the oil temperature measurement has a faster response than the plate centerline temperature measurement to any heat input. Also, the thermocouple used for plate centerline temperature measurement is insulated from the reaction clutch plate because it is held in place by a ceramic cement which is less heat conductive than the steel reaction clutch plate material.

In Figure 13e (as well as in Figures 13a - 13d) after the first few seconds of clutch engagement, the reaction clutch plate temperature becomes greater than the oil exit temperature. Thus convective heat transfer from the

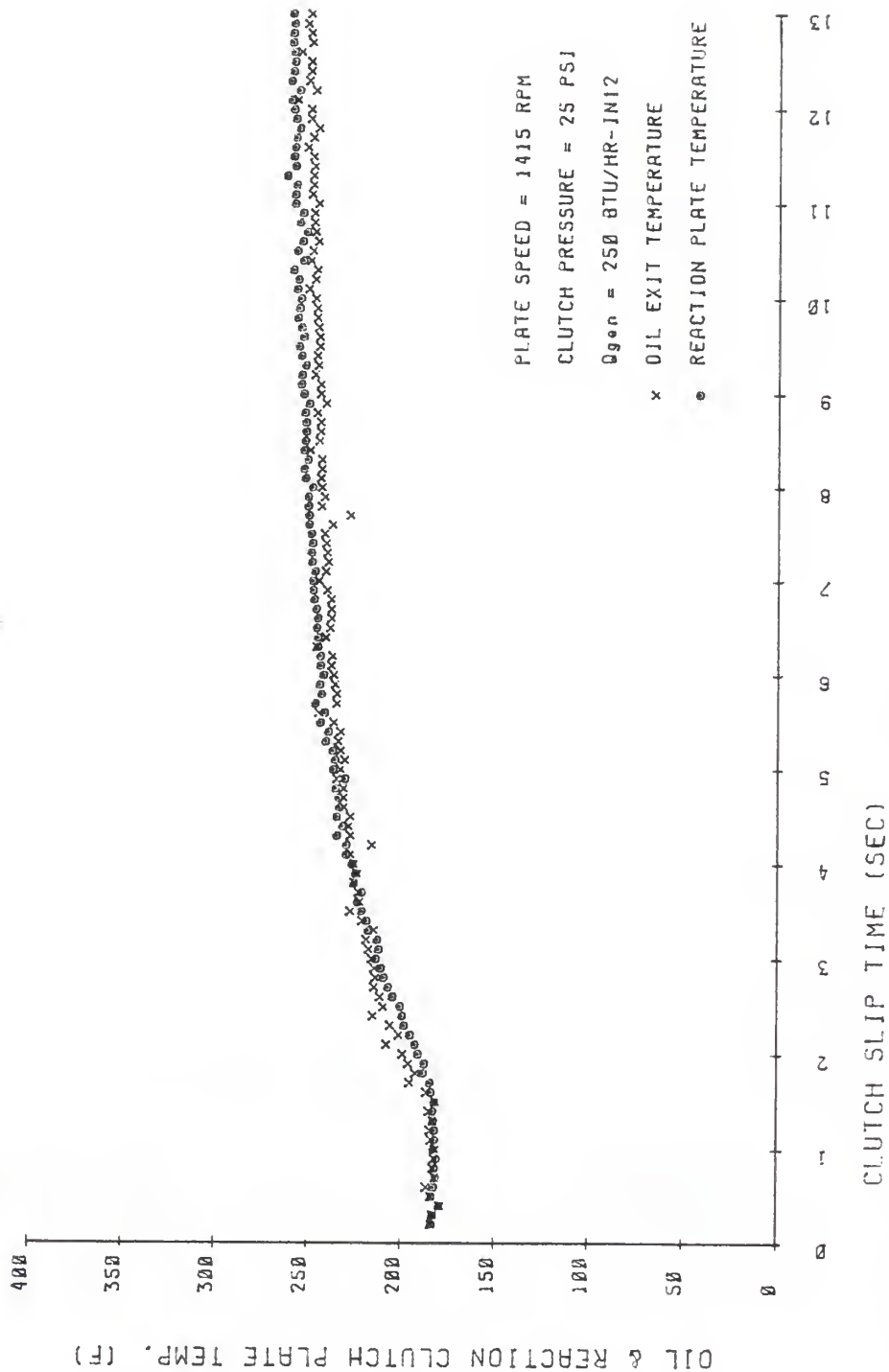


FIGURE 13a. TEMPERATURE PROFILES WITH STEP HEAT INPUT

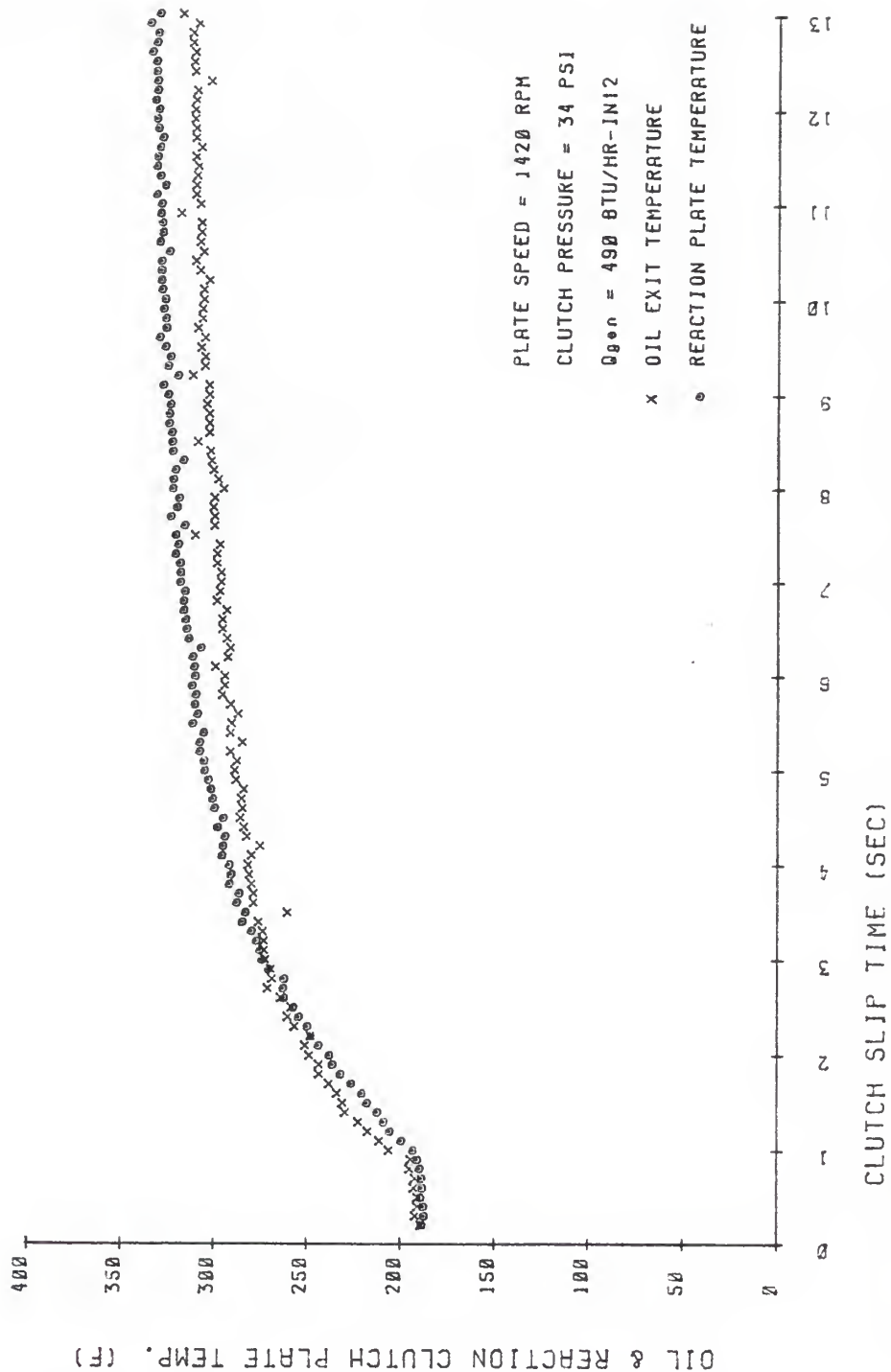


FIGURE 136. TEMPERATURE PROFILES WITH STEP HEAT INPUT

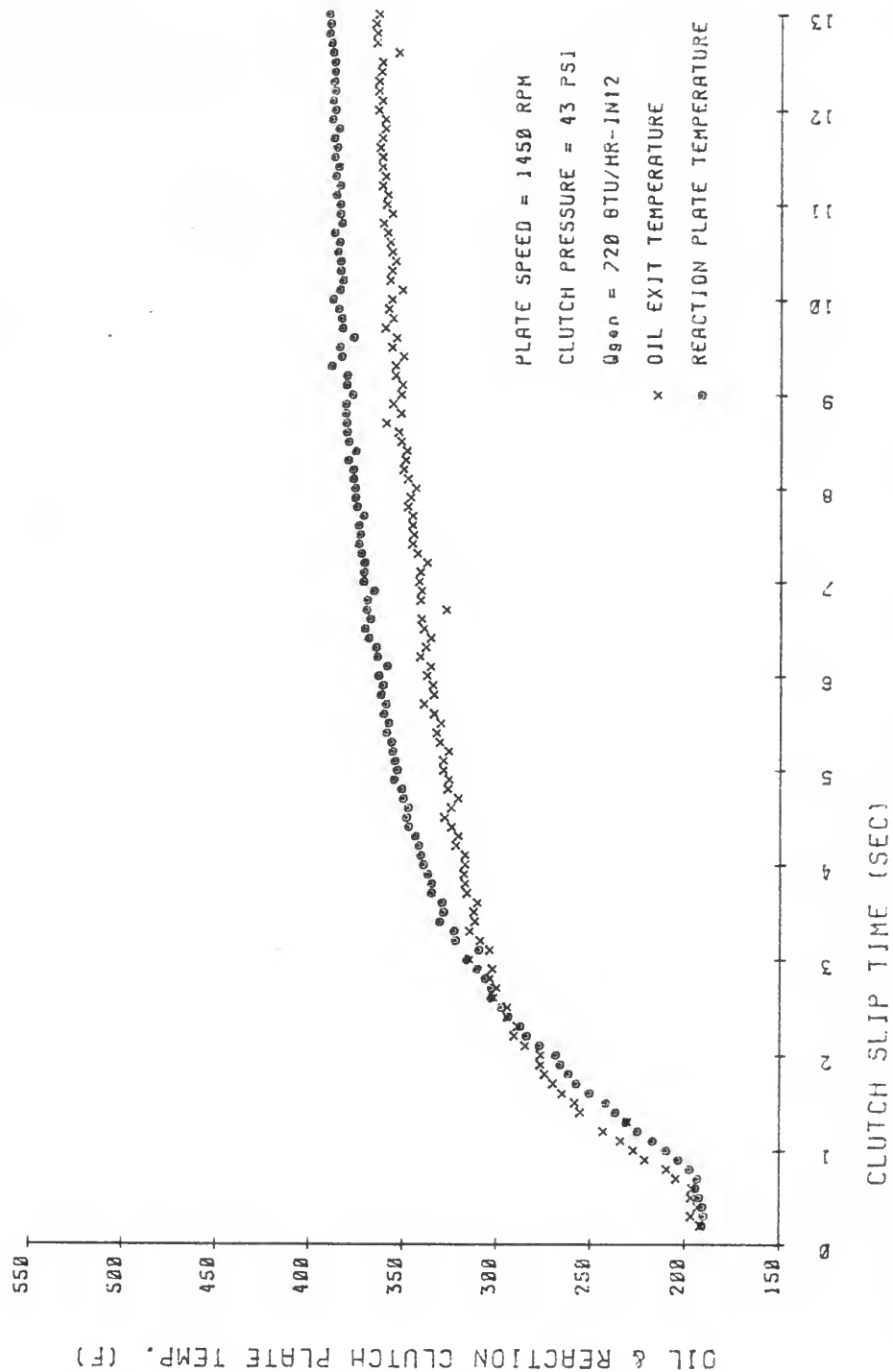


FIGURE 13c. TEMPERATURE PROFILES WITH STEP HEAT INPUT

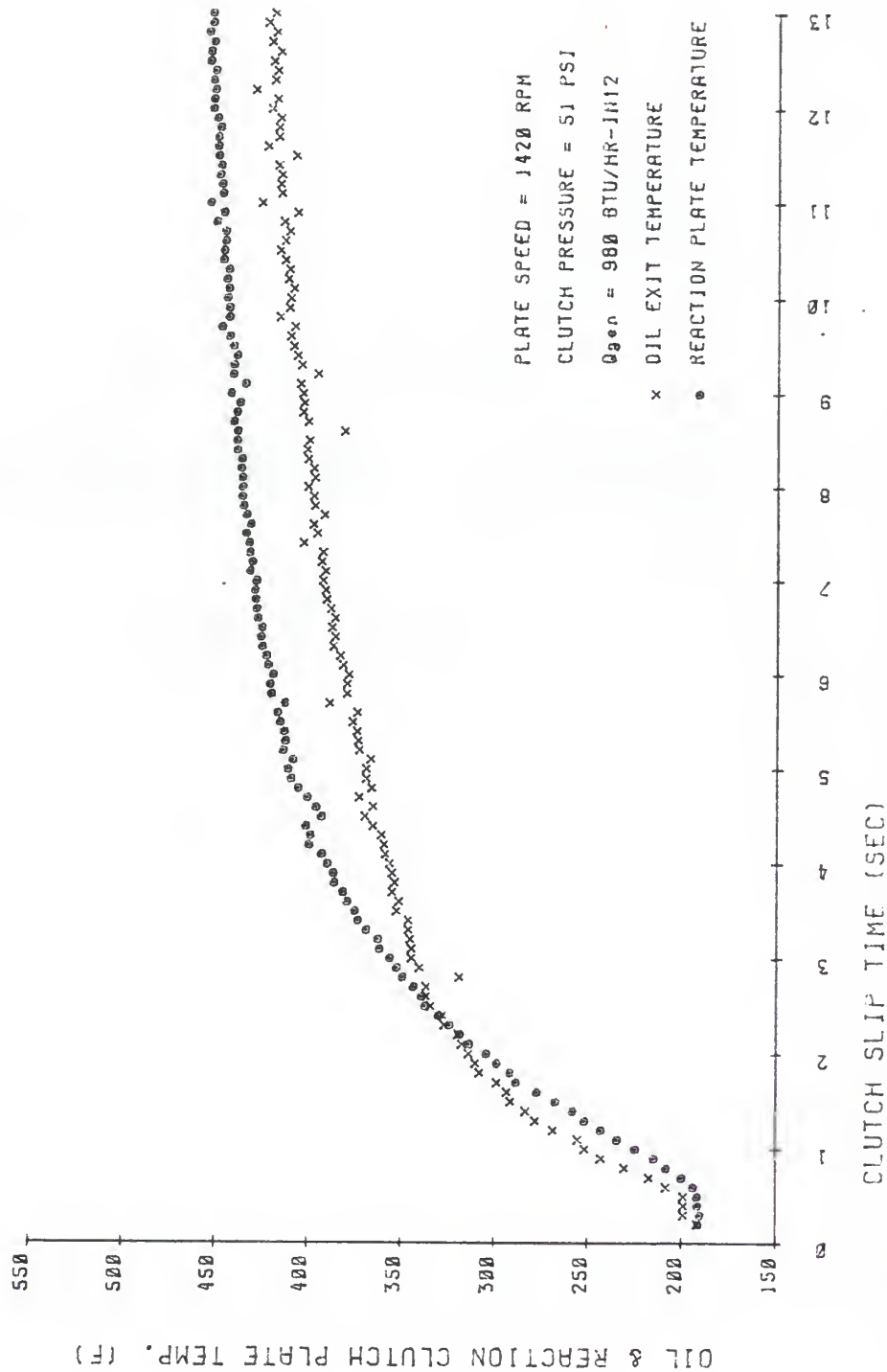


FIGURE 13d. TEMPERATURE PROFILES WITH STEP HEAT INPUT

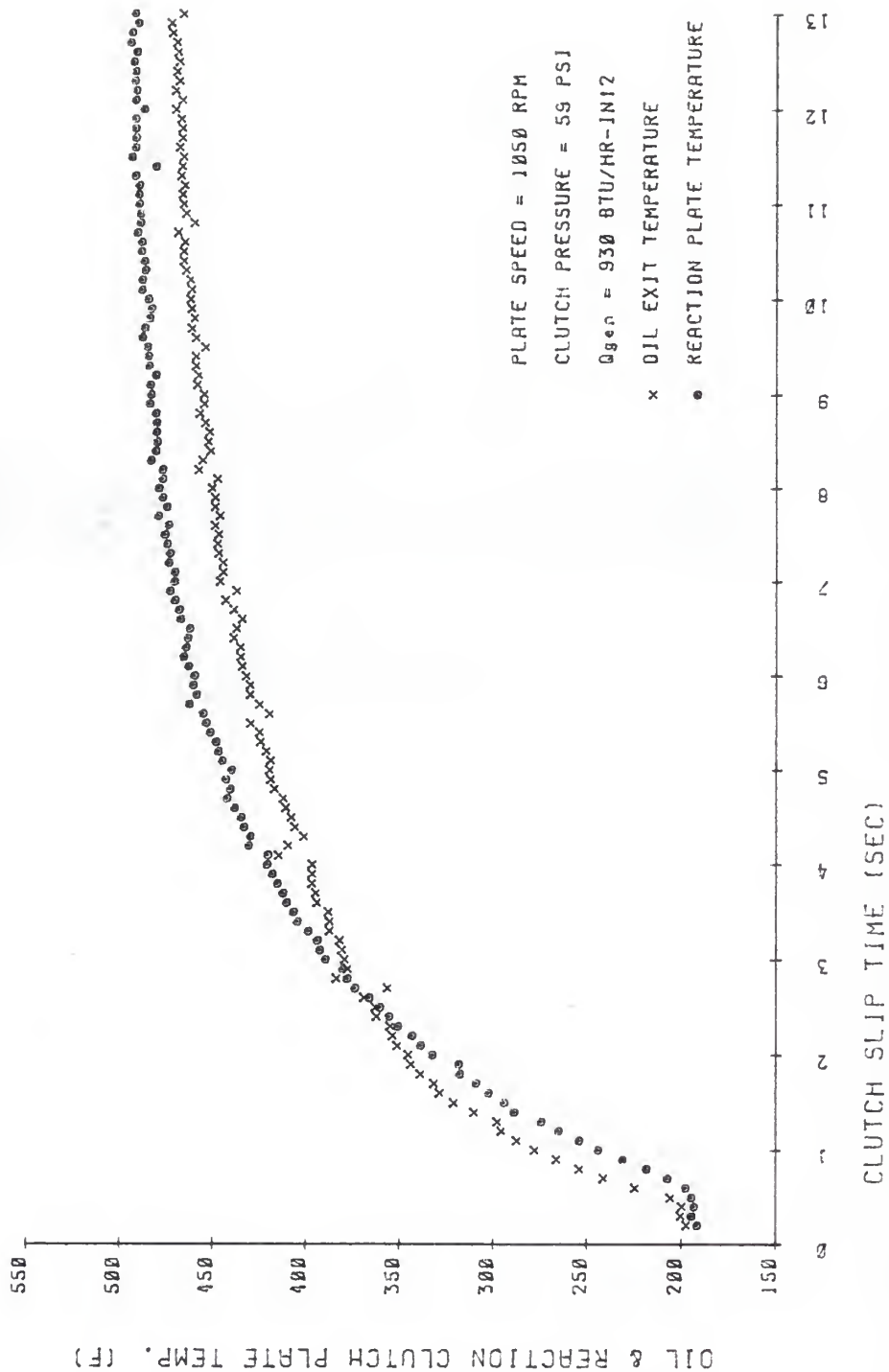


FIGURE 13e. TEMPERATURE PROFILES WITH STEP HEAT INPUT

reaction clutch plate surface occurs because of the temperature difference between the plate surface temperature and the oil temperature. Because the oil carries the input heat out of the clutch, the plate temperature was expected to be greater than the oil temperature.

Clutch Heat Transfer Coefficient

Equation 4 was derived earlier in this thesis using the assumption of a lumped parameter clutch model. However, various other assumptions can be made to determine the relative importance of the different parameters in Equation 4.

If it is assumed that T_{PLATE} is approximately equal to T_{OUT} then Equation 4 becomes:

$$H1 = 16.042 *$$

$$\frac{GPM [RHO(T_{OUT}) C_p(T_{OUT}) T_{OUT} - RHO(T_{IN}) C_p(T_{IN}) T_{IN}]}{A (T_{OUT} - T_{IN})} \quad (6)$$

Or, if it assumed that T_{OUT} is approximately equal to T_{PLATE} then Equation 4 becomes:

$$H2 = 16.042 \text{ GPM} *$$

$$\frac{[RHO(T_{PLATE}) C_p(T_{PLATE}) T_{PLATE} - RHO(T_{IN}) C_p(T_{IN}) T_{IN}]}{A (T_{PLATE} - T_{IN})} \quad (7)$$

If T_{IN} is approximately the same temperature as T_{PLATE} (this

is true for an open clutch and also for slip times close to zero in an engaging clutch) then the oil properties may be factored out of Equation 6 to yield Equation 8 below.

$$H_3 = \frac{16.042 \text{ GPM } \rho(T_{OUT}) C_p(T_{OUT})}{A} \quad (8)$$

The equations above assume that the reaction clutch plate centerline temperature is equal to the reaction clutch plate surface temperature. Heat transfer then occurs due to the temperature difference between reaction clutch plate surface temperature and oil fluid temperature. Since the heat generation (proportional to torque times speed) occurs at the plate surface, the heat will be convected away from the plate surface by the oil. Thus it can be expected that the reaction clutch plate temperature will be noticeably higher than oil exit temperature. Equation 4 contains no assumptions other than a lumped clutch model. Equations 6-8 contain different assumptions in order to determine the effects of those assumptions by comparing various computed heat transfer coefficients. Thus the various parameters which affect the heat transfer coefficient may be examined more closely.

Heat transfer coefficients for the clutch were calculated using Equation 4 and Equations 6 - 8. These equations used experimentally measured values of clutch oil flow rate, oil inlet and outlet temperatures and reaction clutch plate

temperatures.

As discussed previously, the reaction clutch plate temperature used for the calculation was measured using a "thick" plate and the axial plate location chosen was at the center of the clutch pack (reaction clutch plate 5). The plate temperature was measured using six thermocouples inserted into the circumference of the plate (see Plate 4). The plate temperature value used in the heat transfer calculation was the average of these six plate temperatures. The temperature of the oil exiting the clutch was measured using a "thin" reaction clutch plate located at plate position 6 (plate position 1 is next to the clutch piston). This plate was rotated so that the thermocouple was at circumferential position 2. The drilled hole method of oil exit temperature measurement was the method used for all such oil exit temperature measurements.

The thermocouple wires were threaded out of the clutch (see Plate 11), through the shroud (see Plate 12) and out of the transmission. The sealed bearing was installed, and then the shroud, discussed previously, was placed around the clutch cylinder to collect the oil flowing through the clutch. A turbine flowmeter was placed in the shroud oil exit line to measure clutch oil flow rate. Figure 11 shows how the oil flow rate changes as the clutch pack closes and then engages. A step change in heat input into the clutch plates begins upon clutch engagement (the actual heat input

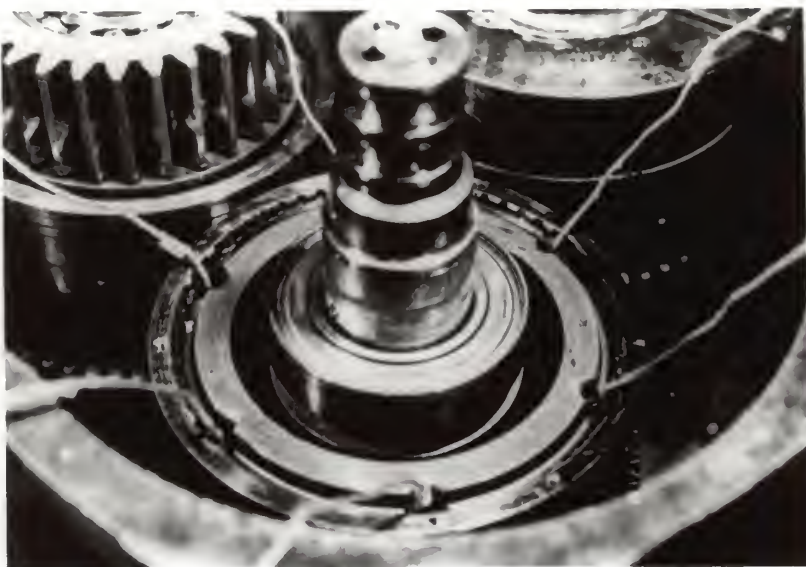


Plate 11. Thermocouple Leads Threaded Out of Clutch

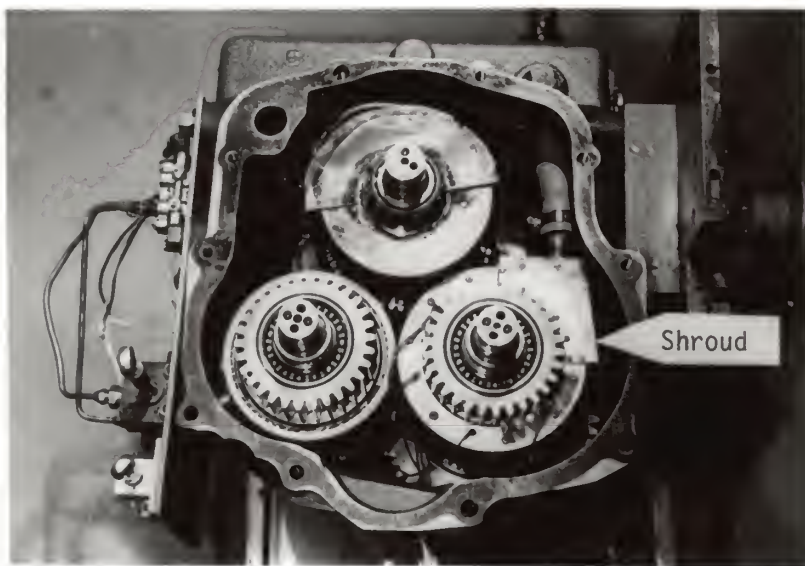


Plate 12. Thermocouple Leads Threaded Through Shroud

is not a perfect step change since the clutch piston is pressurized over a finite interval of time).

Calculated values for three of the heat transfer coefficients defined (H , H_2 , and H_3) are shown as functions of time in Figures 14a - 14e. These coefficients were calculated using Equations 4, 7 and 8, respectively. The heat transfer coefficient H_1 is not shown in these figures because the results for H_1 are nearly identical to those for H_2 . Equation 4 is the basic heat transfer coefficient equation, and Equations 6 - 8 each contain assumptions that simplify the heat transfer coefficient expression. This was done to determine which of the parameters dominate or are temperature sensitive in Equation 4. Note that in Figure 14a (as well as in Figures 14b - 14e) all of the heat transfer coefficients have the same shape as the oil flow rate through the clutch, previously shown in Figure 11a. The parameters involving temperature in Equation 4 simply shift the heat transfer curves up or down. Thus it can be concluded that the oil flow rate through the clutch is the most important single factor involved in the heat transfer coefficient calculation. In fact, if the oil flow rate and the initial reaction clutch plate temperature were the only known conditions within the clutch, a computer model could accurately predict the reaction clutch plate temperatures (this would be the same as using H_3 , Equation 8, for the heat transfer coefficient).

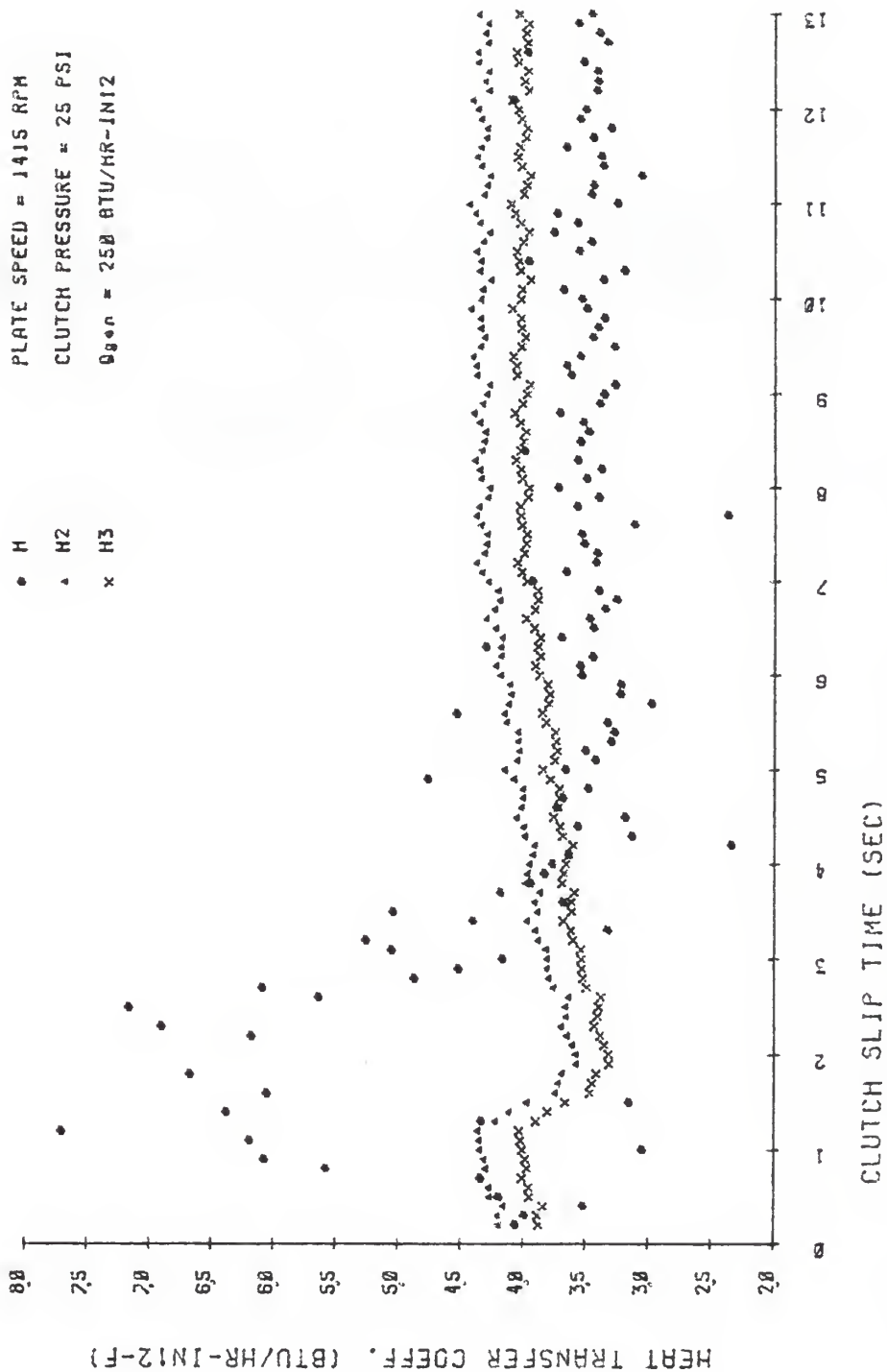


FIGURE 14a. EXPERIMENTAL HEAT TRANSFER COEFFICIENTS

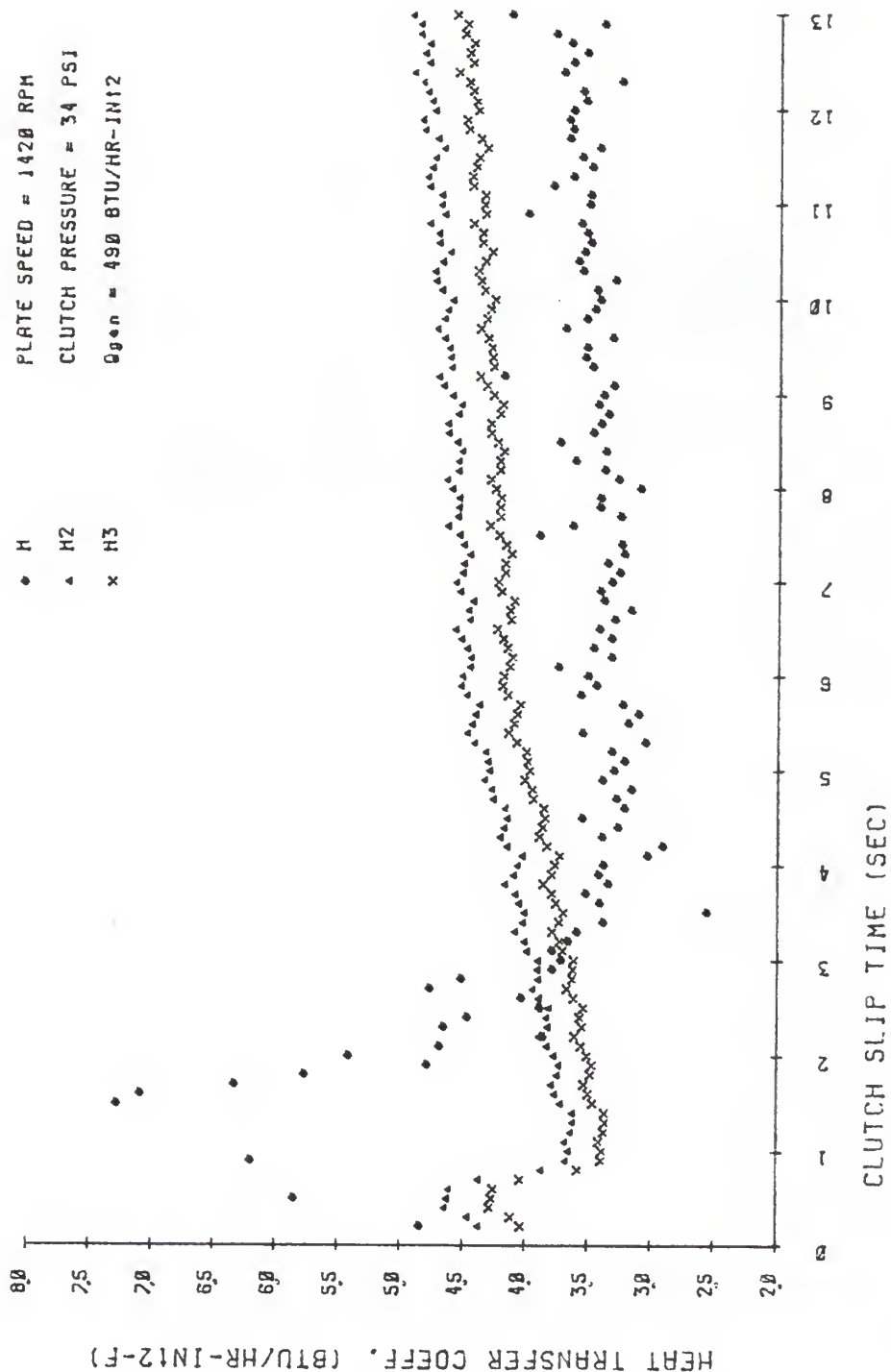


FIGURE 146. EXPERIMENTAL HEAT TRANSFER COEFFICIENTS

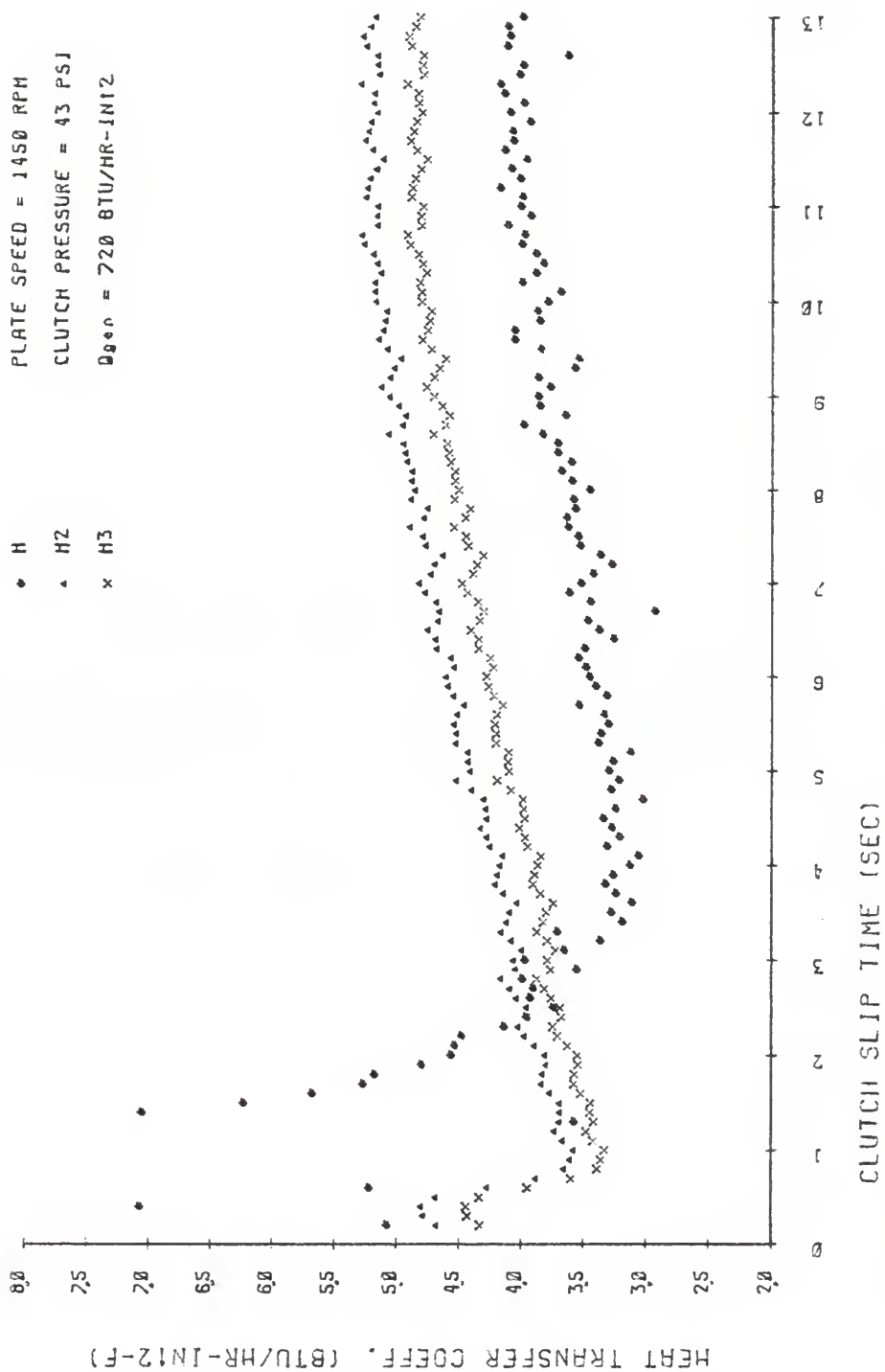


FIGURE 14c. EXPERIMENTAL HEAT TRANSFER COEFFICIENTS

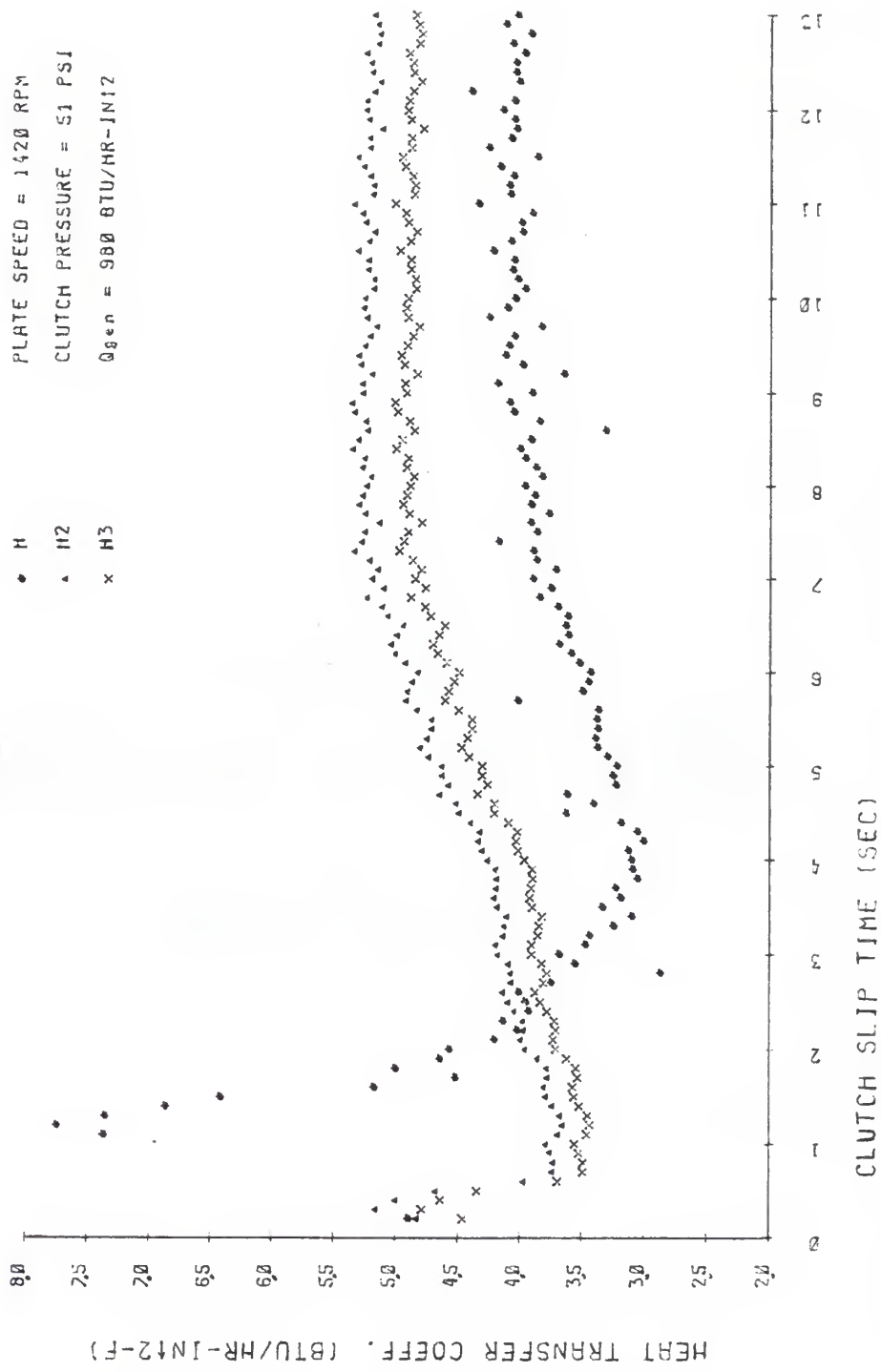


FIGURE 14d. EXPERIMENTAL HEAT TRANSFER COEFFICIENTS

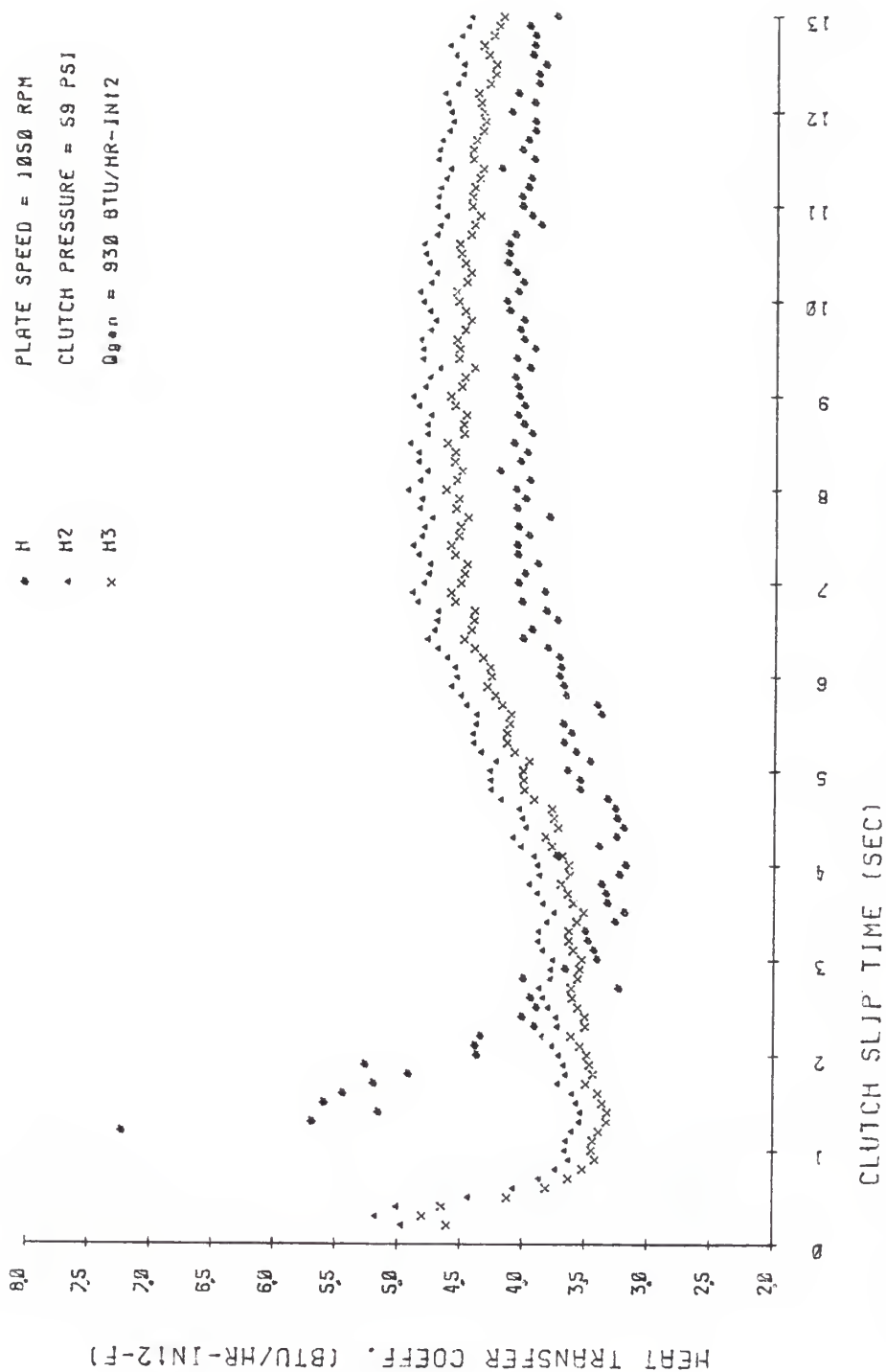


FIGURE 14e. EXPERIMENTAL HEAT TRANSFER COEFFICIENTS

Equation 4, used to calculate the heat transfer coefficient H , does not give satisfactory results during the initial clutch slip time (the first two or three seconds) because of temperature and oil flow rate measurement errors. Because all of the initial temperature values (the temperature values during the first several seconds of clutch engagement) used in Equation 4 are nearly the same, the overall error associated with these temperature measurements becomes larger when they are subtracted and divided as required by Equation 4. It may be recalled that the thermocouple transmitters alone each have an error of 1.5 F. Furthermore, the difference between the temperature of the oil exiting the clutch and the reaction clutch plate temperature ranges from only five degrees to twenty degrees. And since the temperature of the oil entering the clutch remains constant, this means that the difference between the temperature of the oil entering the clutch and the arithmetic mean oil temperature ranges from under 10 degrees when the clutch is initially engaged, to over 200 degrees difference at the end of the clutch engagement. Thus the shape of the heat transfer coefficient curve during this initial clutch slip time should be determined by Equations 6 - 8 instead of Equation 4 (see Figures 14a - 14e). However, Equation 4 does give more accurate heat transfer coefficient values after these first several seconds of clutch slip time (once the temperature differences become large enough to

reduce the amount of error involved in the calculation).

Before testing was begun, the reaction clutch plate temperature was expected to approach a steady state temperature value since the heat input to the clutch remained constant over the clutch slip time. However, the plate temperature values in Figure 13e (and also in Figures 13a - 13d) do not reach a steady state temperature, but rather increase at a constant rate. Therefore, the heat transfer coefficients in Figure 14e (and also in Figures 14a - 14d) should decrease (have a negative slope) with increasing clutch slip time. In order for the heat transfer coefficient to decrease, the oil flow rate through the clutch (shown in Figure 11e) should also decrease, as discussed previously. Note, however, that in Figures 14d - 14e the heat transfer coefficients do finally decrease after a certain maximum point is reached. This decrease occurs when the oil exiting the clutch becomes somewhat stabilized with regard to the amount of air entrainment and oil vapor within the oil flow. It is believed that this negative slope is indicative of what accurate values for the experimental heat transfer coefficient should be. This decrease in flow as a function of clutch slip time is shown in Figures 11d - 11e.

In order to improve predicted heat transfer coefficient values, it may prove desirable to include in Equation 4 such factors as the engine speed and torque converter speed

ratio (or clutch plate speed), and clutch piston pressure (or heat generation). Furthermore, the gear train efficiency was not accounted for in the torque measurement. An efficiency of 98% would likely be satisfactory. This factor is not entirely negligible since 2% of a maximum heat generation of 1400 btu/hr/in² could raise the computed plate temperature by several degrees. However, none of these factors are directly included in the present model (Equation 4). Also, the distance between the clutch and the turbine flowmeter causes a time lag in the flowmeter response.

CHAPTER VI

PREDICTED REACTION CLUTCH PLATE TEMPERATURES DURING "INCHING" USING TRANSIENT CLUTCH HEAT TRANSFER MODEL

The transient clutch heat transfer model used to predict reaction clutch plate temperatures during "inching" operation is derived below. First, assuming conservation of energy gives:

$$Q_{GEN}(t) - Q_{CONV}(t) = Q_{STORED}(t) \quad (9)$$

Then substituting for $Q_{CONV}(t)$ and $Q_{STORED}(t)$ gives:

$$Q_{GEN}(t) - H A [T(t) - T_{MEAN}(t)] =$$

$$RHO C_p A L \frac{[T(t + \Delta t) - T(t)]}{\Delta t} \quad (10)$$

Finally, substituting for $T_{MEAN}(t)$ (note that this definition assumes that the temperature of the oil exiting the clutch is equal to the reaction clutch plate temperature) and rearranging Equation 10 gives:

$$T(t + \Delta t) =$$

$$T(t) + \frac{\Delta t}{RHO C_p L} \{Q'_{GEN}(t) - 0.5 H [T(t) - T_{IN}]\} \quad (11)$$

where:

- A = total reaction clutch plate surface area, in²
- C_p = specific heat of steel reaction clutch plate, btu/lb/F
- H = heat transfer coefficient, btu/hr-in²-F

L = reaction clutch plate half thickness, in
 $Q_{GEN}(t)$ = heat generated at reaction clutch plate surface at time t , is proportional to torque times speed
 btu/hr
 $Q'_{GEN}(t)$ = heat flux per unit surface area, btu/hr/in²
 $Q_{CONV}(t)$ = heat convected from reaction clutch plate surface at time t , btu/hr
 $Q_{STORED}(t)$ = heat stored by reaction clutch plate at time t ,
 btu/hr
 RHO = density of steel reaction clutch plate, lb/ft³
 $T(t)$ = reaction clutch plate temperature at time t , F
 $T(t+dt)$ = reaction clutch plate temperature at time
 $t+dt$, F
 T_{IN} = temperature of oil entering clutch, F
 $T_{MEAN}(t) = 0.5 [T_{IN} + T(t)]$ = arithmetic mean oil
 temperature at time t , F
 Δt = time increment, sec

A computer program, which solves Equation 11 iteratively, was written to calculate reaction clutch plate temperatures using experimentally obtained heat transfer coefficient values and heat flux values. The computer program, by the forward difference method, uses known quantities at time t to calculate a future reaction clutch plate temperature at time $t+dt$. It was originally planned that the heat transfer coefficients, as functions of time, would be obtained from Equation 4. However, it was previously concluded that the present experimental heat transfer coefficient results (obtained through Equations 4 and 6 - 8) are unsatisfactory (due to oil flow rate measurement errors). Thus the experimentally determined heat transfer coefficients were not used in the computer model. Rather, the heat transfer coefficient values were assumed to be constants, chosen to achieve the same steady state reaction clutch plate

temperatures as was experimentally measured. The heat flux used came from actual experimental data, therefore it was a function of time.

Predicted reaction clutch plate temperatures are compared to measured reaction clutch plate temperatures in Figures 15a - 15e in order to verify the accuracy of assuming that the heat transfer coefficients are constant. Achievement of the same steady state reaction clutch plate temperatures (experimental temperatures and computed temperatures) verifies that the proper heat transfer coefficient value was substituted into the computer model, Equation 11. The assumed heat transfer coefficient value used is shown in each of the figures.

The assumed heat transfer coefficient constants (shown in Figures 15a - 15e) are 25% - 50% greater than the experimental heat transfer coefficients (shown in Figures 14a - 14e). It was expected that the experimental heat transfer coefficient values would be greater than the assumed values because, as was previously concluded, air and oil vapor caused inaccurate turbine flowmeter measurements, thus causing high experimental heat transfer coefficient values.

In Figure 15e, the computed reaction clutch plate temperature is greater (by up to 35 degrees) than the measured reaction clutch plate temperature during the initial portion of the exponential temperature curve. This

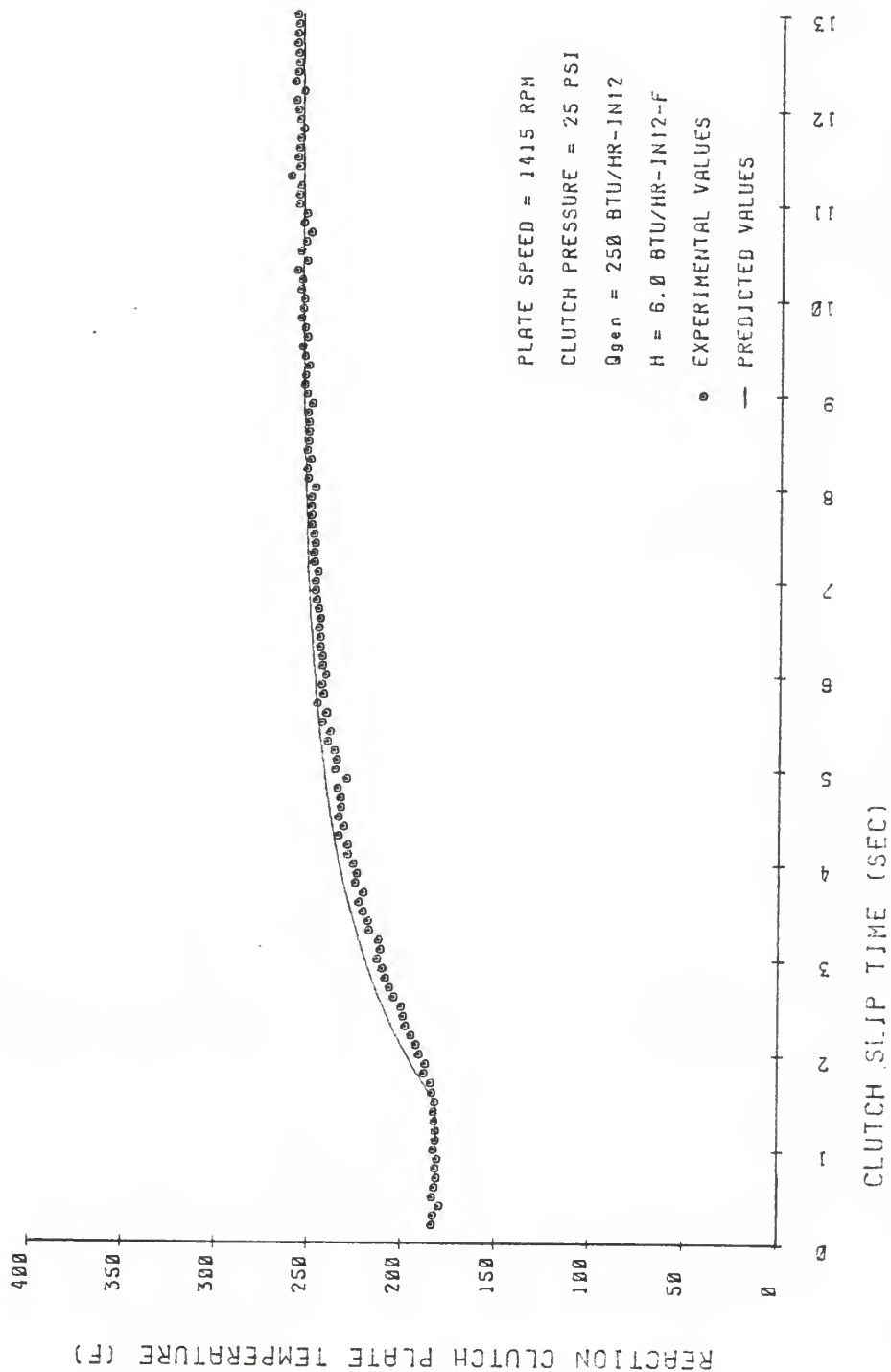


FIGURE 15a. EXPERIMENTAL / PREDICTED REACTION CLUTCH PLATE TEMPERATURES

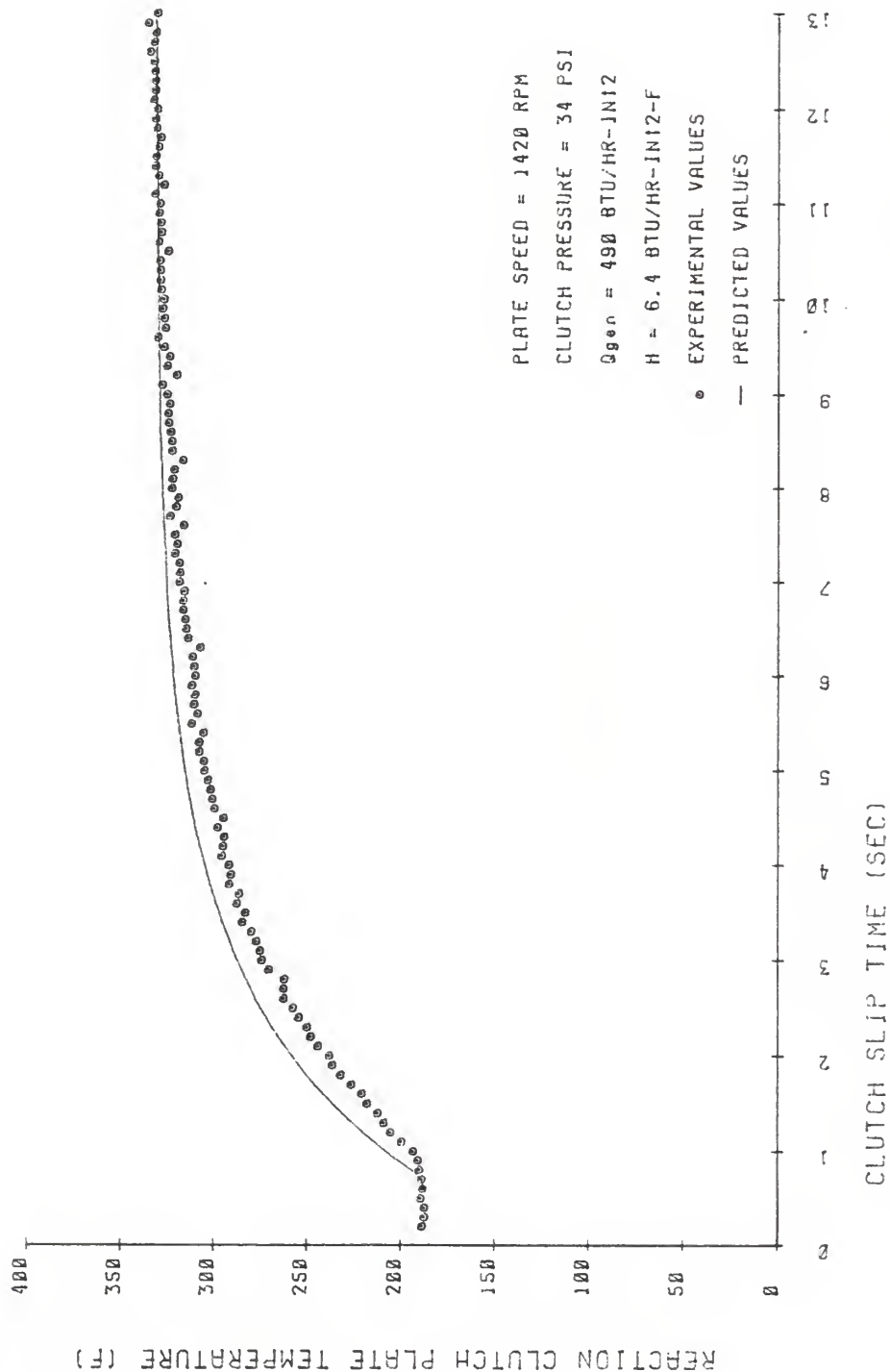


FIGURE 15b. EXPERIMENTAL / PREDICTED REACTION CLUTCH PLATE TEMPERATURES

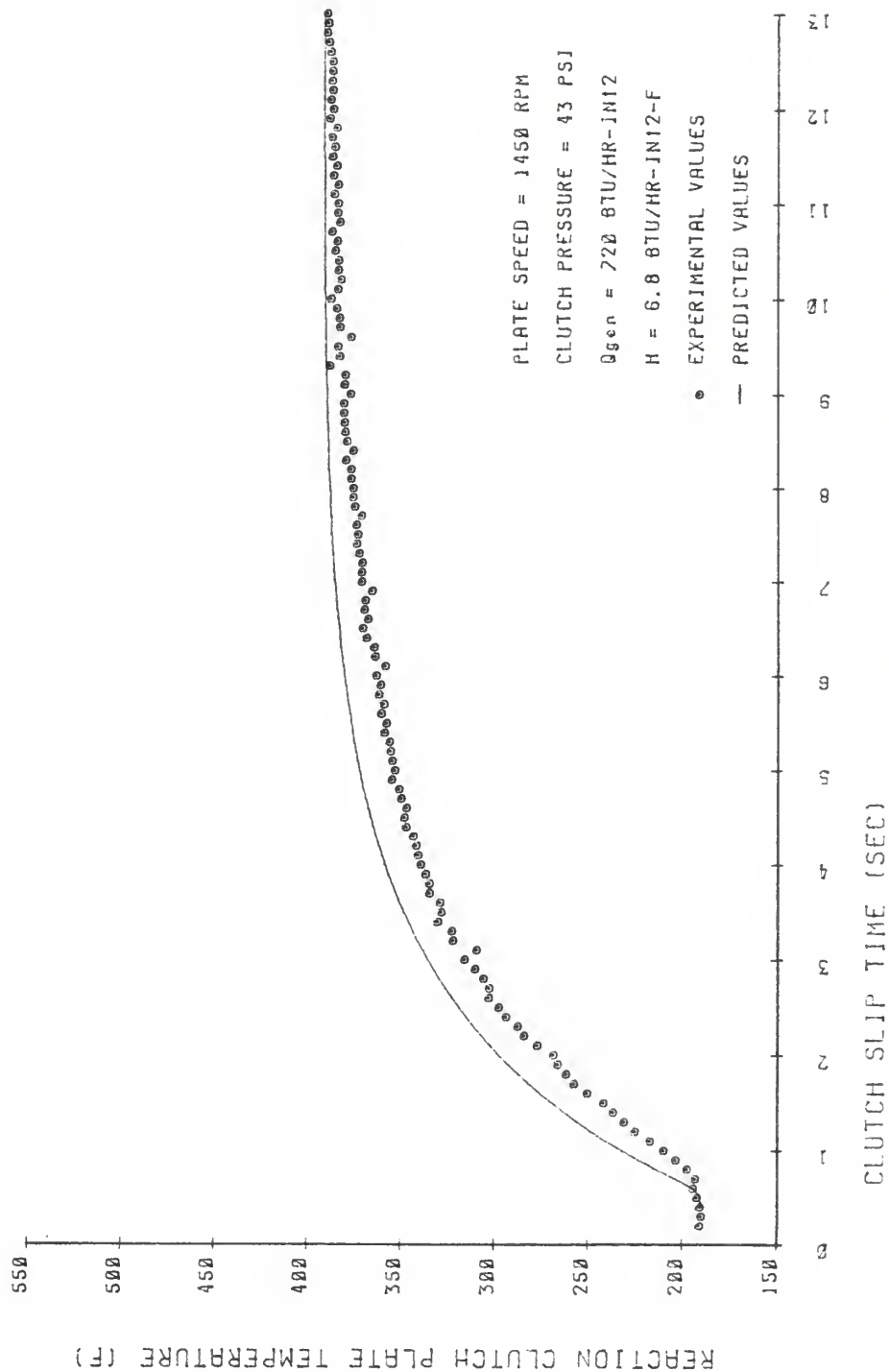


FIGURE 15c. EXPERIMENTAL / PREDICTED REACTION CLUTCH PLATE TEMPERATURES

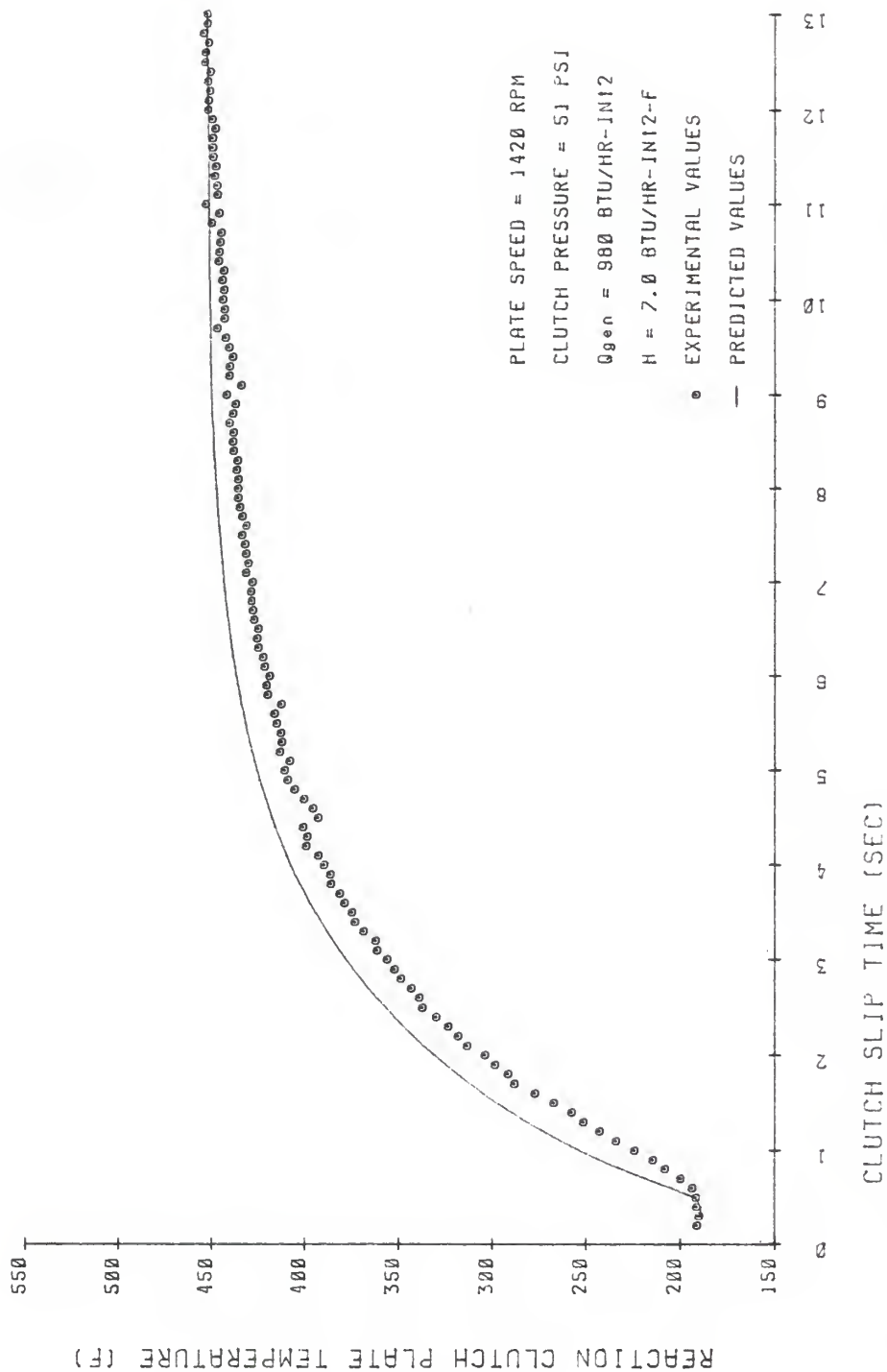


FIGURE 15d. EXPERIMENTAL / PREDICTED REACTION CLUTCH PLATE TEMPERATURES

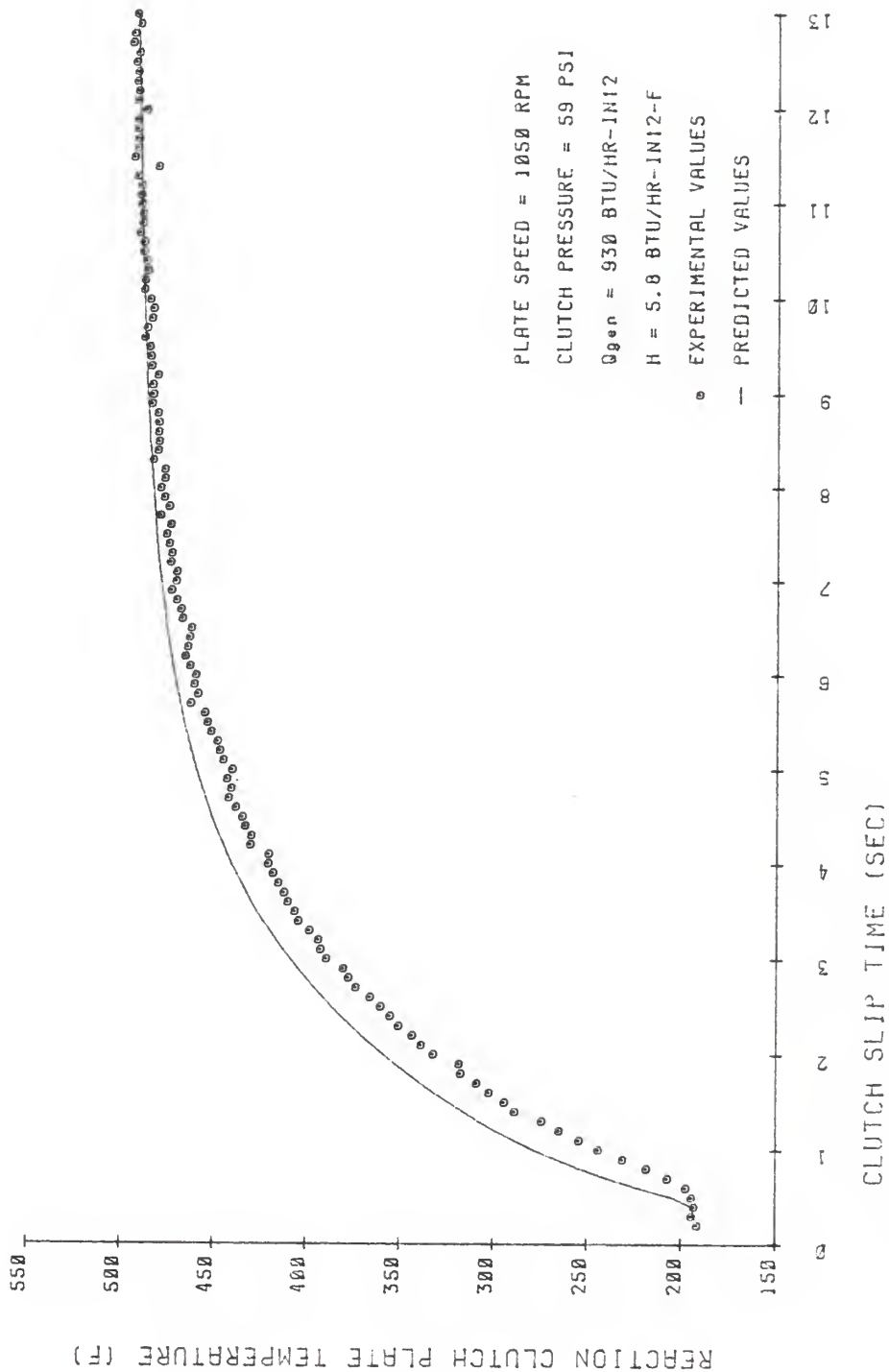


FIGURE 15e. EXPERIMENTAL / PREDICTED REACTION CLUTCH PLATE TEMPERATURES

was expected for three reasons. First, the heat transfer model used does not allow any heat to enter the friction plates, since the friction material is assumed to be an insulator. Second, the thermocouple measuring the reaction clutch plate temperature is held in place by ceramic cement, which has a lower thermal conductivity than the steel reaction clutch plate material. Third, the heat transfer coefficient used in the computer model is a constant, while the experimental heat transfer coefficient is believed to decrease as a function of reaction clutch plate temperature (this is the expected result if the oil flow rate measurement errors could have been corrected). The results shown in Figures 15a - 15e allow evaluation of the experimental heat transfer coefficient values since the experimental results should agree with the computer results. Comparison between the heat transfer values used in the computer model and the heat transfer values obtained experimentally show the amount of error involved in the experimental heat transfer coefficient results. The heat transfer coefficient H (calculated using Equation 4) in Figures 14a - 14e comes the closest to the heat transfer coefficient values used in the computer model (after H reaches its lowest values, several seconds after the clutch has begun slipping). The experimental heat transfer coefficient values would be more accurate, and would even compare favorably to the heat transfer coefficient values

substituted into the computer model, if the experimental heat transfer coefficient values did not increase with time (caused by oil flow rate measurement errors).

The heat transfer coefficients used in the computer model (these heat transfer coefficient values are constants) are shown in Figure 16 for friction clutch plate speeds of 1400 rpm. These results indicate that as long as the friction clutch plate speed remains constant, the heat transfer coefficient is a linear function of clutch piston pressure.

The experimental reaction clutch plate temperatures lag the predicted reaction clutch plate temperatures in Figures 15a - 15e. This error is due in part to the assumption that no heat enters the friction clutch plates. In order to improve the clutch heat transfer model, a second computer program was written using the inverse heat conduction method to calculate the actual heat input into the reaction plates. The inverse heat conduction computer program uses the experimental reaction clutch plate temperatures as input data, and calculates the heat input required to achieve those temperatures. The heat into the friction clutch plates is equal to the total heat generated minus the heat into the reaction plate minus the heat convected to the oil. Thus, through a thermodynamic heat balance around the clutch, the temperature of the friction clutch plates may be computed. The fraction of heat input into the friction plates will be a small percentage of the total heat

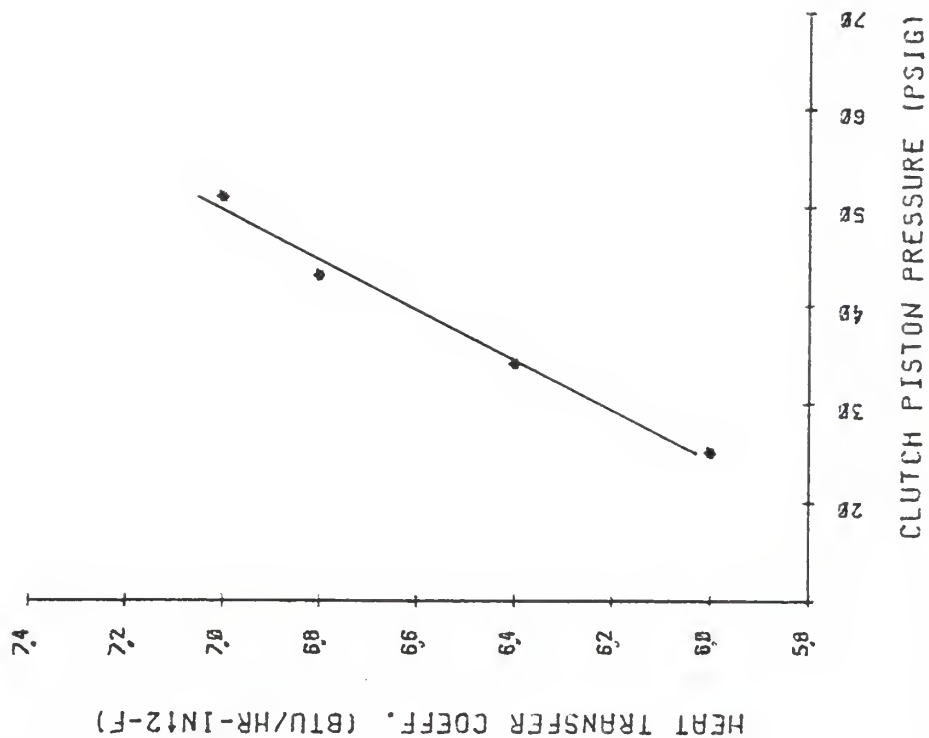


FIGURE 16. PREDICTED HEAT TRANSFER COEFFICIENTS

generated because the asbestos friction material acts as an insulator. This portion of the heat transfer into the friction clutch plate is not entirely negligible [1]. This last step concerning the inverse heat conduction model was not completed because of the inability to adequately determine the oil flow rate through the clutch, which convects the heat away from the clutch.

CHAPTER VII

CONCLUSIONS

1) A test setup has been developed which can be used to study the heat transfer characteristics of the clutches in a powershift transmission.

2) Changing the transmission oil circuit to allow measurement of the oil flow rate upstream from the clutch, rather than downstream from the clutch, as in this research, should eliminate the oil flow rate measurement errors caused by air and vapor in the oil stream.

3) The true oil flow rate through the clutch, during tests which simulated "inching", does not increase with time, but rather is either constant or else decreases with increasing plate temperature.

4) The uncertainty of the measured values of oil flow rate through the clutch during tests simulating "inching" prevented accurate determination of the clutch heat transfer coefficient.

5) The oil flow rate through the clutch has a far greater affect on the clutch heat transfer coefficient than either clutch plate temperature or oil temperature.

6) Accurate determination of the oil flow rate through the clutch should permit accurate calculation of the amount of heat convected from the clutch plates, and thus accurate calculation of clutch plate temperatures during simulated

"inching".

7) The average heat transfer coefficient should decrease with increasing clutch slip time because the reaction clutch plate temperature values increase at a constant rate as time increases.

8) The heat transfer coefficient is a linear function of clutch piston pressure for constant friction clutch plate speeds.

9) There are significant temperature variations, axially, radially and circumferentially, within the clutch that should be considered when measured temperatures are used in heat transfer coefficient calculations.

CHAPTER VIII

RECOMMENDATIONS

1) The oil flow rate should be measured upstream from the clutch where air in the oil is minimal, thus allowing accurate determination of oil flow rate. The shroud can then be eliminated as well as the flow restriction caused by the insertion of a turbine flowmeter downstream from the clutch. Elimination of the shroud is also desirable in that it will facilitate installation of thermocouple leads, and reduce any flow restriction caused by the leads.

2) The temperature variations, axially and radially, within the clutch should be more thoroughly investigated in order to locate temperature measurement points which, when averaged, would give representative temperature values for the overall clutch.

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NOMENCLATURE

a	Reaction clutch plate thermal diffusivity, ft^2/hr
A	Total reaction clutch plate surface area, in^2
C _p	Specific heat of steel reaction clutch plate, $\text{btu}/\text{lb}/\text{F}$
C _p (T)	Specific heat of oil at temperature T, $\text{btu}/\text{lb}/\text{F}$
GPM	Volumetric oil flow rate through clutch, gal/min
H	Heat transfer coefficient, $\text{btu}/\text{hr}-\text{in}^2-\text{F}$
k	Reaction clutch plate thermal conductivity, $\text{btu}/\text{hr}/\text{ft}/\text{F}$
K	Conversion factor
L	Reaction clutch plate half thickness, in
\dot{m}	Mass oil flow rate through clutch, lb/min
Q _{GEN} (t)	Heat generated at reaction clutch plate surface at time t, btu/hr
Q _{GEN} (t)	Heat flux per unit surface area, $\text{btu}/\text{hr}/\text{in}^2$
Q _{CONV} (t)	Heat convected from reaction clutch plate surface at time t, btu/hr
Q _{STORED} (t)	Heat stored by reaction clutch plate at time t, btu/hr
RHO	Density of steel reaction clutch plate, lb/ft^3
RHO(T)	Oil density at temperature T, lb/ft^3
t	Time, sec
T _{AVE}	Average of the axial reaction clutch plate temperature measurements, F
T _{IN}	Temperature of oil entering clutch, F
T _{OUT}	Temperature of oil exiting clutch, F
T _{PLATE}	Reaction clutch plate temperature, F
T _{MEAN}	Arithmetic mean oil temperature, F
T _{MEAN} (t)	Arithmetic mean oil temperature at time t, F
T(t)	Reaction clutch plate temperature at time t, F
T(t+dt)	Reaction clutch plate temperature at time t+dt, F
Δt	Time increment, sec
ΔT _{MAX}	Maximum reaction clutch plate temperature difference, F
ΔT _{MIN}	Minimum reaction clutch plate temperature difference, F

APPENDIX A

MATERIAL PROPERTIES AND DIMENSIONS

Value	Definition
k = 3.41	Thermal conductivity of OMEGA CC cement @ 500 F, btu-in/hr/ft ² /F
k = 25	Thermal conductivity of steel, btu/hr/ft/F
k = 300	Thermal conductivity of steel, btu-in/hr/ft ² /F
RHO = 490	Density of steel, lb/ft ³
RHO = 54.9	Density of oil @ 60 F, lb/ft ³
C _p = 0.11	Specific heat of steel, btu/lb/F
C _p = 0.45	Specific heat of Conoco C3 oil @ 0 C, btu/lb/F
C _p = 0.47	Specific heat of Conoco C3 oil @ 68 F, btu/lb/F
C _p = 0.79	Specific heat of Conoco C3 oil @ 400 C, btu/lb/F
L = 0.035	"Thin" Reaction clutch plate half thickness, in
L = 0.0625	"Thick" Reaction clutch plate half thickness, in
A = 219.5	Total reaction clutch plate surface area, 10 plates, in ²
K = 60	
$\dot{m} = \text{GPM} * \text{RHO}(T) * 2.31 / 1728$	
$T_{\text{MEAN}} = 0.5 (T_{\text{IN}} + T_{\text{OUT}})$	
$T_{\text{MEAN}}(t) = 0.5 [T_{\text{IN}} + T(t)]$	
OMEGA thermocouple wire, HH-K-24	
Borg-Warner SD 1053 clutch plate friction material	

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COMPUTER MODELING OF THE HEAT
TRANSFER IN A POWERSHIFT TRANSMISSION
CLUTCH UNDER SLIPPAGE

by

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AN ABSTRACT OF A MASTER'S THESIS

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requirements for the degree

MASTER OF SCIENCE

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1987

ABSTRACT

The objective of the research presented in this thesis was to develop and experimentally verify a transient heat transfer model which predicts the temperature, as a function of time, in the reaction clutch plates of a Funk 2133 series powershift transmission when "inching" takes place at a constant rate of energy dissipation (heat input).

The test setup developed consists of three main components: a transmission (the object under study), an engine, and a data acquisition system.

Preliminary testing consisted of measuring the oil flow rate through the clutch and measuring temperatures within the clutch. The oil flow through the forward directional clutch was collected by installing a special shroud around the clutch cylinder. The oil flow rate was then measured two different ways: either directly (by measuring time and oil volume) or by turbine flowmeter. The temperature distributions within the clutch were investigated and the reaction clutch plate temperatures were found to change in the axial, radial and circumferential directions. The temperature of the oil exiting the clutch was also measured.

A heat transfer model was developed, and experimental results were used to calculate the clutch heat transfer coefficient. The heat transfer coefficient values and the

heat flux into the clutch were then used in a computer model to predict reaction clutch plate temperatures.

Because of difficulties in measuring the oil flow rate accurately, due to air and oil vapor within the oil, the experimental heat transfer coefficients, and thus the predicted reaction clutch plate temperatures, are in error.

It is recommended that the test setup be modified to permit measurement of clutch oil flow rate upstream from the clutch, instead of downstream as in this research. This should be done before any further research is attempted.